# EFFECTS OF REAR BUMPER BEAM DELETION ON THE PERCEPTION OF STEERING PERFORMANCE OF COMMERCIAL VEHICLES

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**Submitted for the degree of Master of Science by Research**

**Faculty of Engineering and Informatics**
**Bradford Programme of Engineering Quality Improvement**

**University of Bradford**

**2015**

## ABSTRACT

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Title of Thesis:- **Effects of Rear Bumper Beam Deletion on the Perception of Steering Performance of Commercial Vehicles**

In order to remain competitive in the marketplace, all motor vehicle manufacturers face difficult decisions with regard to balancing *cost vs. feature*. That is to say that the manufacturer must balance the cost of the product to the customer to remain competitive whilst offering appropriate technology and standard features required by that customer.

All motor manufacturers are therefore under pressure to keep costs of non-feature items to a minimum. One of the cost reductions items prevalent on most vehicles is the deletion of the structural member that attaches the rear bumper, known as the *bumper beam* (RBB), which is researched in this Thesis. This generates average vehicle savings of $20 and, as this is invisible to the customer, should enable the manufacturers to realise a significant saving or allow this revenue to be spent on additional feature without loss of vehicle function.

However, in nearly all cases, deletion of the rear bumper beam has the effect of degrading the steering responses of the vehicle by 1 to 1½ rating points (out of 10), which is contrary to the premise of cost reductions; which is to ensure that vehicle function is unaffected.

Initial analysis of vehicles with deleted rear bumper beams cannot show an objective measurable difference in any vehicle behaviours with or without the beam fitted, and hence CAE studies using ADAMS models cannot verify the effects of the bumper beam. It was necessary to employ unconventional modelling and testing methods such as rigid body, flexible body model techniques as well as experimental studies included driving robots and expert driver appraisals.

The research demonstrated that vehicle modelling methods currently used, cannot establish or predict the complete vehicle ride and handling status. A total vehicle model approach should be used without separating the body CAE model and vehicle dynamics ADAMS model into separate entities.

Furthermore, it was concluded that the determination to the effects of body hysteresis rather than pure stiffness is of crucial importance and that the steering attribute could be maintained with the deletion of the RBB analytically.

Key Words:- **Steering, Perception, Analytical, Objective, ADAMS, Stiffness, Hysteresis**

Project Supervisors: Prof. Andrew day, Dr. Khalid Hussain

# ACKNOWLEDGEMENTS

The Author wishes to express sincere appreciation to the following people for their assistance to enquiries during the preparation of this project.

University of Bradford
Professor Felicean Campean
Dr. Khalid Hussain
Alan Brunning

Ford Motor Company
Dr. Jeremy Couval
Dr. Armin Lepold
Dries Proost
Koen Bex

After a very motivated start to this project, I was faced with some very difficult hurdles to overcome. I switched jobs which made access to some of the information required to complete this project very difficult to come by. Plus in 2008, the economic downturn in Europe had a catastrophic effect on the resources available to me at Ford Motor Company. This put the completion of the project at all in jeopardy. It was only by the good grace of Jeremy Couval who volunteered to do the ADAMS modelling in his own time that I could even begin to complete this project. My heartfelt thanks go to Jeremy for making this happen!

Finally, I'd like to thanks my family, Lisa, Juliet & Jack, for their encouragement, understanding, good humour and sacrifices made to enable me to commit myself to this Masters course in its entirety.

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# DECLARATION

I declare that the contents of this project, except where otherwise stated, are the results of my own work and have been conducted with the full knowledge of Martina Beyer, Manager European Suspension Applications Manager, Chassis Engineering, Ford Motor Company Ltd.

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Signed: ……………………………

Date: 30th April 2015

# LIST OF FIGURES

# GLOSSARY OF TERMS

| Term | Description |
| --- | --- |
| ADAMS | Automated Dynamic Analysis of Mechanical Systems |
| BIC | Best In Class |
| BIW | Body In White |
| CAD | Computer Aided Design |
| CAE | Computer Aided Engineering |
| CoG | Centre of Gravity |
| DNA | Deoxyribo Nucleic Acid |
| DOE | Design of Experiments |
| DVP | Design Verification Plan |
| FEA | Finite Element Analysis |
| FMEA | Failure Mode and Effect Analysis |
| FRT | Front |
| FWD | Front Wheel Drive |
| G | Gravity |
| K&C | Kinematics and Compliance |
| MATLAB | Matrix Laboratory |
| NCAP | New Car Assessment Program |
| OEM | Original Equipment Manufacturer |
| RBB | Rear Bumper Beam |
| RR | Rear |
| RWD | Rear Wheel Drive |
| SAE | Society of Automotive Engineers |
| SDS | System Design Specification |
| SLD | Side Load Door |
| SWA | Steering Wheel Angle |
| TGW | Things Gone Wrong |
| USG | Understeer Gradient |
| VER | Vehicle Evaluation Rating |
| VW | Volkswagen |

# NOMENCLATURE

| Symbol | Description | Units |
|---|---|---|
| $A_y$ | Lateral Acceleration | $°/s^2$ |
| $C_{af}$ | Tyre Force on Front Axle | N |
| $C_{ar}$ | Tyre Force on rear Axle | N |
| E | Young’s Modulus | Pa |
| F | Force | N |
| $f_n$ | Natural Frequency | rad/s |
| g | Gravity | $m/s^2$ |
| $I_{xx}$ | Moment of Inertia in *X* Direction | $kg/m^2$ |
| $I_{yy}$ | Moment of Inertia in *Y* Direction | $kg/m^2$ |
| $I_{zz}$ | Moment of Inertia in *Z* Direction | $kg/m^2$ |
| K | Understeer Gradient | °/g |
| M | Mass | kg |
| $W_f$ | Mass of Front axle | kg |
| $W_r$ | Mass of Rear Axle | kg |
| V | Velocity | m/s |
| X | Displacement in Longitudinal Direction | mm |
| Y | Displacement in Lateral Direction | mm |
| Z | Displacement in Vertical Direction | mm |

# 1. INTRODUCTION

## 1.1 Background of Vehicle Handling

Early days of vehicle design and manufacture were in the main, concerned with engine development, specifically engine power, to make cars achieve higher speed (Gillespie, 1992). Vehicles had poor handling, ride comfort, the steering was frequently unstable and the front axle and brakes made poor performance almost inevitable (Olley, 1957). It quickly became apparent that to achieve greater speed on various road surfaces, a greater emphasis and study would be required on the braking and steering capabilities of the vehicle. Hence, ride and handling analysis have become important in vehicle design studies.

Vehicle dynamics is the science of vehicle suspension systems. The primary function of this is to isolate the vehicle structure (and occupants) from vibrations from the road surface. This is referred to as *Ride*. The suspension must do this whilst maintaining stability, body control and overall driver confidence. This is referred to as *Handling* However, it is often the case that ride and handling requirements conflict and so detailed engineering studies are required to achieve the best compromise (Howard, 1987).

It is worth noting that the term *handling* is often used interchangeably with cornering, turning or directional response (Gillespie, 1992). However, there are subtle differences between these terms. Cornering, turning or directional response are objective attributes of the vehicle during changes of direction, and are quantifiable.

For example, directional response is determined by the time required for lateral acceleration and yaw rate to develop following a steering input. This is often referred to as *transient response*. By definition, lateral acceleration and yaw rate are measurable responses.

In contrast, handling is the subjective feeling fed back to the driver in order to make the driving characteristics of a particular manoeuvre more confident, which affect the drivers' ability to maintain control of the vehicle. This implicates not only the explicit vehicle capabilities, but also the importance of the vehicle-driver interaction.

## 1.2 Vehicle Dynamic Understanding & Measurement

The empirical understanding of vehicle handling is based on trial and error. That is to say, no mechanical understanding is used to predict results of design changes (Gillespie, 1992). The empirical handling aspects of a vehicle are important, as these are the impressions felt by the driver. Good driver feedback can give the impression of a quality product, even from a vehicle with average handling characteristics (Johnson, 2005).

### 1.2.1 Empirical Research

The empirical method of understanding vehicle handing will often lead to failure, or at best create very long periods of development. Although empirical development can be applied with success if experienced engineering experts are given a well defined target (Metz, 2004).

### 1.2.2 Analytical Research

To understand and make use of the laws of physics, an analytical understanding of how vehicle properties affect the handling of a vehicle is required to bridge the gap between what is felt by the driver and what is known by the engineer (Crolla et al, 1998). The method and approach to translating the voice of the customer into engineering requirements is always the most difficult part of vehicle development, as customer requirements rarely give an engineer enough information to design or develop a system (Burn, 1990; Woehler, 1997).

### 1.2.3 Measurement

Current analytical studies are mainly computer based and describe the mechanics of a system so that an analytical model can be established. The analytical model can then be used to establish the important factors of a system and may be used to predict behaviour if changes occur in the models' foundations. These must be measurable in order to define the models correctly and to quantify outputs should inputs change.

The basis for any analytical model is robustness. It is imperative to ensure that the basic inputs that affect the ability of the model to analyse and predict behaviour under certain conditions, is maintained. Without a robust, correlated model, engineers will not be able to obtain results with any integrity, which in extreme cases may lead to catastrophic results (Banks, 2004). It is therefore of paramount importance that engineers' understand the base assumptions that have been made in the model to avoid such errors.

### 1.2.4 Business Needs of Accurate Model Integrity

The need for accurate model analysis and measurement is ever increasing due to the competitive nature of the automotive business. The goal of all companies is to achieve lower costs whilst improving products that achieve higher function, and to do this in a timely manner. This puts a reliance on accurate analytical modelling as there are often faster in allowing engineers to sign-off changes to products without the need for prototypes or expensive physical or rig based testing.

## 1.3 Current Trends

With the current market demands for low cost, manufacturers are always seeking different methods of lowering costs to remain competitive in the marketplace. These can come from:-

1. Taking advantage of manufacturing and technology advances
2. Reducing the costs of made or bought components, and by utilising low cost manufactures
3. Reducing the costs of made or bought components by limiting profit margins of internal organisations or suppliers
4. Deletion of systems or components that are invisible to the customer
5. Deletion of systems or components that are visible to the customer
6. Revisions to designs that are invisible to the customer (ie. deletion of paint on components out of direct vision – Pedal Boxes etc.)
7. Price revisions to the customer with no contributions from the vehicle systems or components
8. Commonisation programs that increase volume to reduce fixed costs (Clarke, 2004)

A manufacturer would ideally like a component or system that could be made at a low cost without compromising standards of quality. However, the amount of effort required to enable that producer to produce components or systems to the required standard is often extensive and hence expensive.

There is also the possibility of customer backlash if, for instance, a vehicle, system or component is known to be made by a low cost producer purely because they *are* a low cost producer. For example, if BMW were to manufacture vehicles in Pakistan, it is unlikely their vehicles could still command a premium selling or that their customer base would remain solid (Slater, 2003).

Manufacturers in the main will therefore lean towards improving or deleting systems that are invisible to the customer as these will lower cost with no downstream implications. There are instances however, when a system deletion is not thought to have customer impact, does become visible. It is this phenomenon, and the dilemma of deleting a function (invisible to customer or otherwise), that will be investigated in depth in this thesis.

## 1.4 Rear Bumper Beam Deletion

To reduce the low speed impact damage of a vehicle, bumper beams are fitted at the front and at the rear of vehicles to absorb the energy of sub 5mph impacts without causing damage to the underbody or rear end structure. The bumper beams are mounted beneath the cosmetic bumper and are therefore invisible to the customer.

As a cost reduction exercise, the deletion of the front and rear bumper beams was investigated. Whilst the front bumper beam is known to aid front crash performance, the rear bumper beam was thought to be suitable for deletion as modern plastic rear bumpers can receive 15kph impacts without the use of the beam. However, it was noticed that when the rear bumper beam (RBB) was deleted, the perceived steering feel performance of the vehicle deteriorated during a standard lane change manoeuvre (Sluimer, 2003), thus making the deletion noticeable or *visible* to the customer (Esser, 2003). The deterioration metrics will be discussed in later chapters.

Despite the high cost of the RBB, the decision was made not to delete this component due to the steering perception deterioration. Therefore the cost saving potential was not realised.

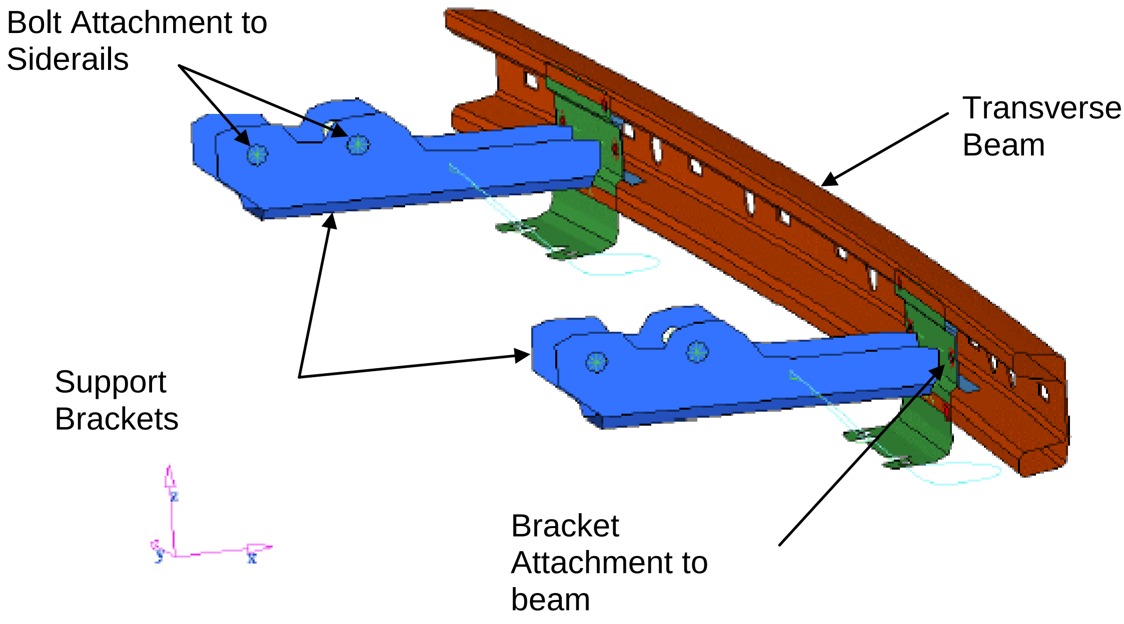



**Figure 1.1** – Rear Bumper Beam Assembly

The RBB is made up of 3 separate fabricated steel components, two support brackets which are bolted to the underside of the vehicles' siderails and the main transverse beam which is bolted to the support brackets as shown in Figure 1.1.

The cosmetic bumper is riveted to the rear end structure of the vehicle and masks the RBB from view.

## 1.5 Aims of Thesis

This thesis investigates the dynamic effects of RBB deletion and how the driver of the vehicle observes changes in dynamic behaviour when the RBB is deleted. It also suggests a suitable alternative to the RBB in order to enable the RBB deletion without degrading vehicle performance. The following will be investigated:-

- The drivers perception of steering feel with and without RBB fitment (Customer trials and questionnaire)
- Physical vehicle dynamic effects of a vehicle with and without RBB fitted and methods of measurement
- Analytical vehicle dynamic effects of a vehicle with and without RBB fitted.
    - Methods
    - Outputs
    - Tyre Models
- Why the RBB plays such a significant part of vehicle dynamic behaviour
- Understanding of the mechanism
- Potential resolution to steering degradation on the vehicle at lower cost that the RBB deletion

The research also only concentrated on the transient steering effects (lane change) of the steering as this *is the issue* reported as functionally degraded.

An in depth literature review will be conducted in order to establish methods and procedures of vehicle design with regard to vehicle measurements, RBB deployment and generic vehicle dynamics related subjects.

Analysis of a vehicle that can be demonstrated to show steering effects with and without an RBB will be completed to confirm results of work done to date, in order to establish a robust baseline to plan further work and establish the link to the customer perception and interface.

The procedure will take four distinct phases which are investigated in depth in later Chapters:-

- Foundation Research – Establishing current knowledge and suggesting future direction
    - Establish customer perception of steering issue (blind, double blind and open)
    - Literature review of related work
    - Critical review of work done to date
    - Brainstorm of possible vehicle responses that could result in steering anomalies
- Physical Research – Investigation of a vehicle with a known RBB deletion / steering link
    - Instrumentation and analysis of an affected Transit van
    - Detailed analysis of measurements and results from investigation
    - Comparison of results analysis and work done to date
    - Confirm degradation can be revalidated (turned on and off)
- Analytical Research – Investigation of a vehicle with a known RBB deletion / steering link to compliment physical research
    - Identified steering response degradation and replicate on analytical model
    - Use analytical model to suggest possible vehicle responses that may cause the identified steering responses
    - Revise analytical model to overcome steering response degradation
    - Develop new measures of steering response
    - Correlate findings with analytical model
- Resolution Research – Correlation of analytical and physical research
    - Direct potential resolutions identified analytically on the vehicle and vice versa
    - Establish best fix based on cost, ease of application, weight, investment and customer perception

## 1.6 Thesis Outline

Chapter 1 has outlined the issue that will be investigated in depth in this thesis. It has also provided an overview of history of vehicle dynamics.

A review of appropriate literature to demonstrate understanding of the subject area is shown in Chapter 2. This shows how vehicle handling is perceived by customers and why it is necessary for direct links between empirical and analytical evaluation methods. The literature review also details the known facts surrounding the subject area and investigates their relevance to the issues contained in the thesis.

In Chapter 3, the history of the vehicle development with and without RBB is outlined with regard to steering and handling effects. Discussions are made around the apparent lack of physical data to support the empirical results and areas of investigation are highlighted for further research that could lead to satisfactorily linking both evaluation methods. These also have links back to CAE tools to ensure correlation of the customer, physical and analytical effects of steering behaviour.

Chapter 4 takes data from the vehicle parameter investigation to analyse if any synergies can be determined that would affect transient handling and the effects of the RBB presence. It suggests methods of combining previous testing and theory work to understand how the RBB deletion could influence steering perception and suggest methods of overcoming this.

Chapter 5 summarises the results of the investigation and testing and draws conclusions from the findings. These are summarised and recommendations for further work in this subject are suggested.

# 2. LITERATURE REVIEW

## 2.1 Introduction

As the RBB deletion has affected subjective yaw response during lane change manoeuvres, areas of investigation can be established. The following chapter reviews literature in related areas:-

- Body stiffness / hysteresis and influences to vehicle handling
- Handling

The RBB deletion is and its effects on handling are currently only quantifiable by subjective evaluation. Investigations will be made into driver perception and its influences in accurate vehicle appraisal.

Influence of stiffness to vehicle systems in the dynamic mode will also be investigated. These factors are:-

- Non-independent suspension systems
- Independent suspension systems
- Tyres & tyre dynamics
- Steering
- Kinematics & Compliance

Methods of experiments of vehicle handling will also be assessed and evaluated to enable different approaches of testing to be established. These will include:-

- Vehicle handling models (Analytical)
- Experimental vehicle handling (Physical)

## 2.2 Body Stiffness & Influences to Vehicle Handling

As steering loss in the form of excessive yaw overshoot has been perceived during lane change manoeuvres, it is probable that a decrease in body stiffness, following the deletion of the RBB, has adversely affected the vehicles ability to react to roll. Lane change manoeuvres by definition occur in the transient domain, therefore body stiffness and its effects on handling must be investigated and understood.

### 2.2.1 Dynamic Requirements of Body Stiffness

Increased body stiffness and rigidity has increased in importance to vehicle quality (Braham, 1997). This also has an effect on the dynamic properties of a vehicle as the suspension can react better to a rigid body than to a more flexible body (Kang et al, 1991).

Body stiffness has been considered recently, more important than strength, as it is possible that a body structure can be sufficiently strong but be unsatisfactory due to insufficient stiffness (Happian-Smith, 2002). Stiffness is ultimately determined by the acceptable limits of deflection of major areas of the vehicle such as door openings, powertrain mountings and occupant interfaces, such as floor and seat panels (Pawlowski, 1969).

Investigations into racing vehicles, has shown that handling balance is strongly influenced by the stiffness of the body. During cornering, the weight transfer between the front and rear axles will determine the amount of oversteer / understeer of the vehicle. The load transfer distribution can therefore only be controlled if the body is sufficiently stiff to transmit the torque (Deakin et al, 2000).

In order to tune vehicle handling, the body must have a torsional stiffness of *X* times the suspension stiffness (Deakin et al, 2000).

Two analysis methods have been developed for body stiffness ability to transmit load transfer:-

- Static model
- Dynamic handling model

Static models can be used to calculate the body torsional stiffness effects on achieving a desired handling balance.

Dynamic handling models can be used to predict the effects of body torsional stiffness on dynamic handling manoeuvres. Dynamic handling model analysis strongly suggests that stiffer bodies require less roll stiffness distributions to achieve the same handling balance. This ratio of chassis torsional stiffness must be a multiple of total suspension roll stiffness and not the difference between front and rear suspension stiffness as has often been suggested (Kang et al, 1991; Deakin et al, 2000).

These studies show that in racing vehicles, the required torsional rigidity required in the body is in the range of 1/2 - 2/3$^{rds}$ of the total roll stiffness. This study has not been completed for road cars or commercial vehicles and must be determined analytically in Chapter 3.

Studies into the effects of global body stiffness and body structure hysteresis have suggested that it is hysteresis and not overall stiffness that has the biggest effect on handling / steering perception (Makino, 2004). This study concluded that reducing the stress concentration at high stress areas such as spotwelds, reduced hysteresis by as much as 30%. Following this study, all Mazda vehicles have a target of less than 5% hysteresis with an overall torsional rigidity target of *competitive* rather than *class leading*. Mazda suggest that rigidity feel correlates to hysteresis but not to stiffness.

The example cited in the Mazda study suggested that the 2004 VolksWagen Polo has lower than average torsional rigidity, but its very low body hysteresis (4.7%) allows a very good rigidity feel.

It is possible that the RBB deletion, whilst having minimal effect on overall torsional stiffness, has degraded the body hysteresis by a larger amount. No data is available for commercial vehicle hysteresis at present and will be investigated in Chapter 3.

### 2.2.2 Handling

Handling describes behaviour of a vehicle to inputs given by the driver. Handling can be classed as subjective and objective, as vehicle behaviour is quantifiable by study but acceptance of this behaviour is ultimately determined by the driver (Gillespie, 1992). Handling falls into 2 categories:-

- Steady state
- Transient

#### 2.2.2.1 Steady State Handling

A steady state exists when vehicle responses to control or disturbance inputs do not change over time. The responses in motion are therefore steady state responses. Steady state responses can act on a vehicle in a straight line, constant radius turn or cambered road (Dixon, 1991; Gillespie, 1992).

It is documented that steady state handling falls into two categories (Dixon, 1991):-

1. Total vehicle handling characteristics (yaw, steer, lateral acceleration)
2. Prediction from individual system designs (suspension, steering etc)

As has been established in Chapter 1, the steering perception loss is noticed during standard lane change manoeuvres, and as such is not affected by the steady state definition. This will be investigated as transient handling.

#### 2.2.2.2 Transient Handling

A transient state exists when motion responses relative to the vehicle or control positions change with time (Gillespie, 1992). By definition, the steering wheel angle must be constantly changing (Metz, 2004). Transient response is the undefined criteria by which drivers refer to good or bad handling (Jacobson, 1983; Metz, 2004).

The transient response time required from the beginning of an input to the percentage of steady state transition is not clearly defined and varies depending on the analysis. Percentage of final steady state transition varies between 50% and 90% (Whitcomb et al, 1956; Bickerstaff, 1976; Pacejka, 1986)

Empirically, the study of transient response is to measure the vehicle response to inputs, and describe the results in terms of characteristics of those particular responses (Barter, 1976). However, this subjective approach does not give consistent feedback to the standard that most vehicle designers can reliably quantify.

Control theory methods developed for the aircraft industry have been adapted successfully for automotive use. It has been established that a 3 degree of freedom model of motion, could predict yaw and roll rate, as well as lateral acceleration frequency responses to steer input (Segel, 1956).

Practical applications of transfer functions on automobiles have been performed successfully and will be investigated in depth in Chapter 3 (Weir et al, 1966; Szostak et al, 1967).

## 2.3 Driver Perception

As the RBB deletion is highlighted as a subjective degradation, investigations into driver perception and its worth in vehicle development must be understood.

Vehicle handling can be objectively measured in great detail analytically and physically (Mashadi et al, 1996; Data et al, 2002; Leonardo et al, 2002; Metz, 2004). Establishing a link between objective measure and driver perception is not an exact science.

Many studies have been undertaken to correlate subjective and objective handling (Abe, 1980; Crolla et al, 1998; Crolla et al, 2000; Martin et al, 2002; Enache et al, 2004; Chai et al, 2005).

Investigations into lane change prediction and driver perception concluded that most drivers show accuracy and consistency when confronted with the same manoeuvre on different vehicles. However, between different drivers, the prediction is less certain. Driver ranking for different vehicles also concluded that driving style is largely independent of vehicle type (Data et al, 2002)

Driver perception studies have shown the necessity to establish fundamental basic and unambiguous criteria for assessment. Many drivers cannot differentiate one engineering attribute from another, therefore analysing results and cross linking to objective data can be misleading (Data et al, 2002).

Whilst it is recognised in general that driver perception and objective measurements can be linked, analytical studies have so far not been developed sufficiently to predict driver reaction adequately (Crolla et al, 2000).

The biggest step forward in linking objective and subjective handling is in the frequency response mode although it is recognised that further work is required before unequivocal links are established (Crolla et al, 1998).

In contrast, some investigations have concluded that subjective ratings for certain events can be predicted using computational models. Mathematical models for subjective lane change prediction have been established and used in some studies. These models are claimed to correlate with yaw response time and undamped natural frequency. It is further suggested that the models are available for use during the engineering development phase of vehicle engineering for theoretical handling predictions (Abe, 1980). There are however, no qualification records to this claim.

Most development on vehicle dynamics use trained, dedicated expert drivers to critically evaluate vehicles. This has advantages such as:-

- Consistent feedback
- Higher standards of driving skill
- Subjective assessments can be made during high lateral acceleration events
- Safety

However, there are also limitations with expert drivers. These are:-

- Feedback can be based on engineering judgement rather than pure subjective feel
- Different ratings may be given when no difference is felt based on a need to find a difference
- Expense

Numerous studies have been completed in an attempt to correlate subjective and objective data, some of them successfully (Chen et al, 1997).

All studies completed so far have studied subjective ratings to a known and quantifiable reference vehicle. Of particular interest in the case of RBB deletion is that sufficient vehicle differences cannot be quantified objectively, therefore subjective correlation cannot be achieved. Chapter 3 will investigate if hysteresis, mentioned in Section 2.2.1., can be differentiated subjectively.

## 2.4 Vehicle Systems

Influence of body stiffness on the suspension performance may have caused inconsistencies from the sign-off position when the RBB was deleted. It is possible for the effects on vehicle systems to be assessed independently, but it is also necessary to analyse the effects from a total vehicle standpoint.

The suspension layout of the subject vehicle is different front to rear as per most light truck applications (Howard, 1987; Dixon, 1991; Gillespie, 1992; Reimpell et al, 1996; Bastow et al, 2004):-

- Rear Suspension – Non-Independent via Hotchkiss suspension
- Front Suspension – Independent via McPherson strut

### 2.4.1 Non Independent Suspension

The Hotchkiss non-independent (or solid axle) suspension is characterised by the wheels being mounted to a rigid beam such that the movement on one wheel is transmitted to the opposing wheel, such that steer and camber angles remain unchanged on the axle, Figure 2.1 (Dixon, 1991; Gillespie, 1992; Bastow et al, 2004).

The advantages of solid axles are (Gillespie, 1992):-

- Wheel camber unaffected by body roll (negligible variation)
- Wheel alignment maintained during movement
- Low tyre wear
- Cost

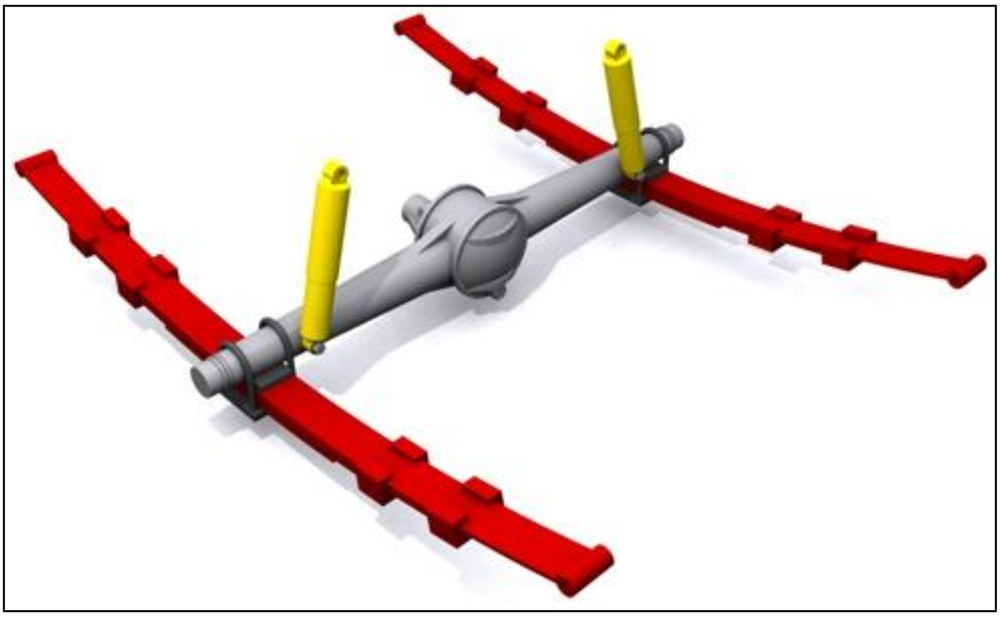


**Figure 2.1** – Hotchkiss Suspension Arrangement (Longhurst, 2006)

Advantages such as these have made solid axles predominant on commercial and off road vehicles (Crolla et al, 1992; Gillespie, 1992). Solid axles do have disadvantages however:-

- Spring rates and efficiencies are affected by bush stiffness
- High unsprung mass
- Anti-roll bars required to be separate item
- Wheel tramp & shimmy common

The effects of reduced body stiffness on the rear suspension in isolation are that during a transient manoeuvre, the weight transfer coefficient will be reduced. As such, the loaded outer wheel will lag the steering input. This is perceived as a lack of steering response by the driver (Gillespie, 1992). During sinusoidal manoeuvres such as accident avoidance, the grip of the rear tyres can lag the front by as much as 70°, Figure 2.2. Longer wheelbase vehicles such as vans, exaggerate this condition.

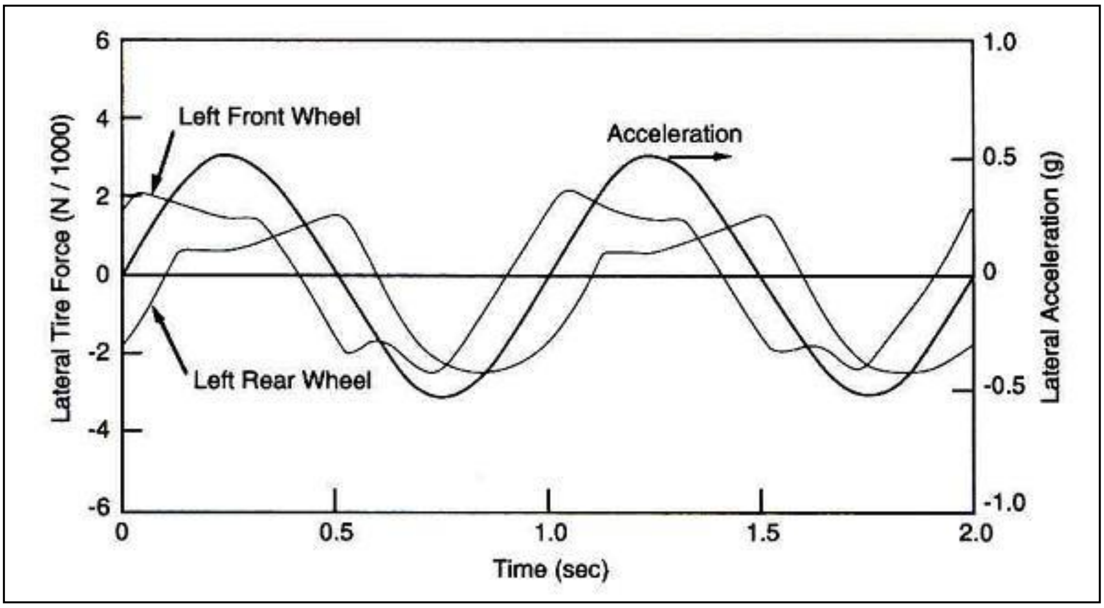



**Figure 2.2** – Tyre Force Phasing vs Lateral Acceleration

However, as tyre phase lag differences are an effect of vehicle handling behaviour rather than a cause, its effects will not form part of the study.

### 2.4.2 Independent Suspension

The McPherson strut independent suspension, Figure 2.3, allows one wheel to move without affecting the opposing wheel (Dixon, 1991; Gillespie, 1992; Bastow et al, 2004).

Most passenger car and light truck application now employ independent front suspension due to their inherent dynamic advantages (Gillespie, 1992).

These advantages are:-

- Compact systems allow package space for powertrain systems
- Steering vibrations are damped
- Larger wheel deflection can be achieved for greater ride comfort
- Greater roll stiffness can be achieved for any given wheel rate
- Roll centre height and instantaneous centre can be tuned in the basic suspension geometry

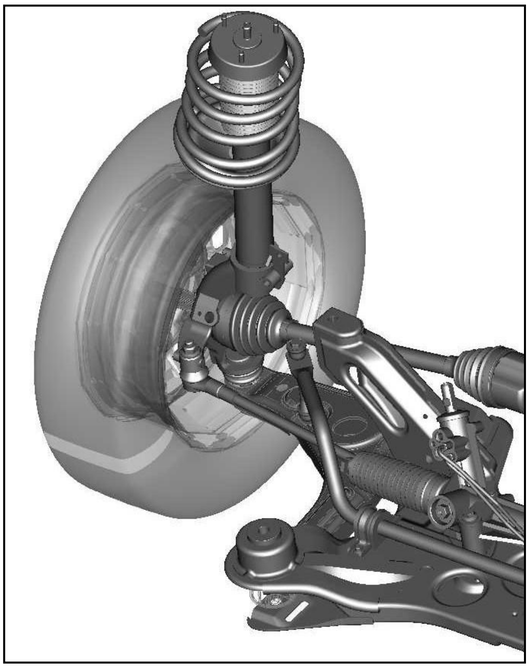


**Figure 2.3** – McPherson Strut Suspension (Banks, 2004)

Disadvantages of independent suspensions are (Gillespie, 1992):-

- Cost
- Infinite variability

As the RBB by definition mainly effects the rear of the vehicle, the lateral weight transfer coefficient reduction due to deletion is only likely to have a negligible effect on the front suspension. Here, the effect will be reduced rear to front weight transfer.

This could potentially mean that the front loaded tyre of the vehicle reacts to less vertical input and therefore undergoes a reduced understeer co-efficient (Naudé et al, 1993). The effect to the driver would be sharper turning response and could exaggerate the phase lag between the front and rear tyres even more. This will be investigated in Chapter 3.

### 2.4.3 Tyres & Tyre Dynamics

Tyre response to vehicle stiffness is inherent as vertical load applied to the tyre is converted to grip. Load transfer generated by body response to steering input produces vertical tyre loads. Tyre grip, adhesion, under load is not a linear correlation, Figure 2.4. That is to say, as the load on a tyre increases, the increase in cornering force generated does not increase by the same amount (Milliken, 1995).

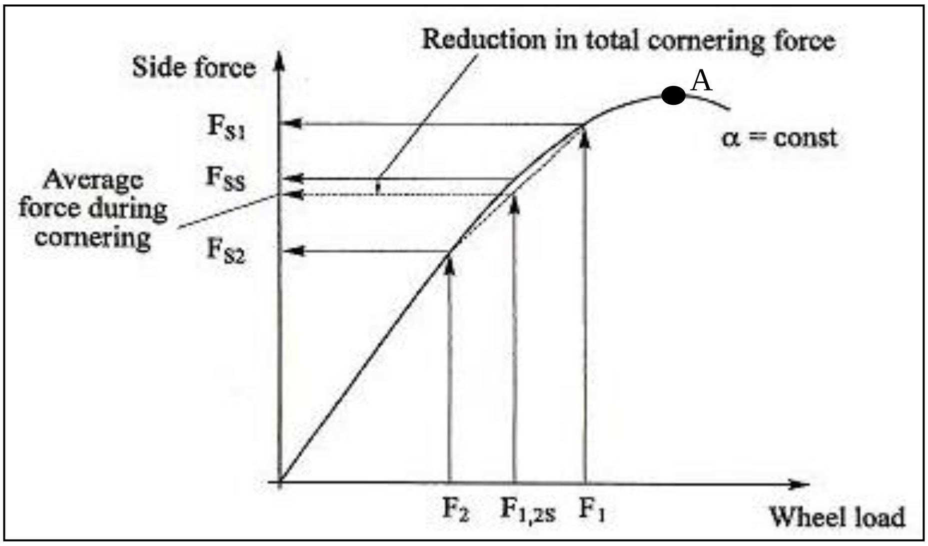



**Figure 2.4** – Weight Transfer Effects on Total Cornering Force (Schalz, 2004)

The non linear behaviour of the tyre performance is due to tyre dynamics. As the vehicle rolls on its roll axis, the tyre contact profile changes as the sidewalls deform and the ultimate contact patch deteriorates, Figure 2.5 (Birch, 1999). Grip will increase on a tyre until it becomes saturated, at which point, the tyre will slide on the road surface, point A in Figure 2.4.

During cornering, the wheels on one axle of a vehicle have different loadings applied due to body roll. The outside tyre is under more load than the inside tyre due to weight transfer effects.

As can be seen in Figure 2.4, the inside tyre, $F_2$, would generate side force $F_{S2}$. The outside, or loaded, tyre, $F_1$, would generate side force $F_{S1}$. Taking a point midway linearly between $F_{S1}$ and $F_{S2}$, it can be seen that during cornering, the average grip of an axle reduces compared to the theoretical average if no cornering forces were present on the axle

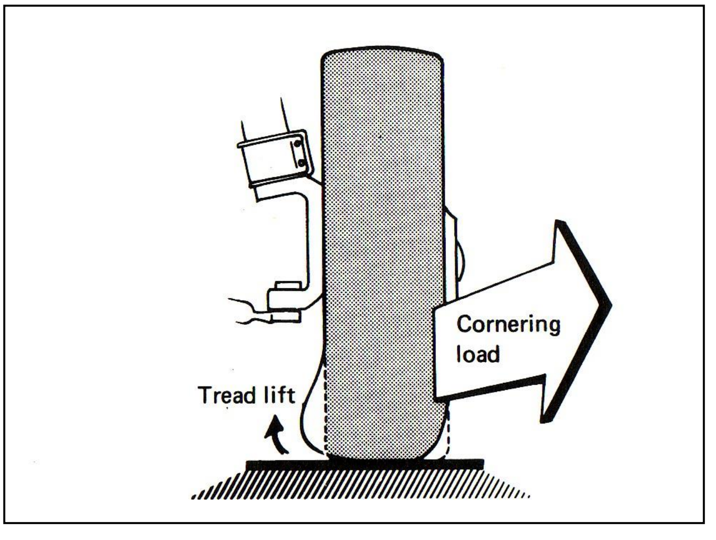



**Figure 2.5** – Tyre Contact Change During Cornering (Birch, 1999)

Another influence of the vehicle during transient manoeuvres is the effect on the dynamic weight distribution due to track change. This does not affect the rear suspension as Hotchkiss suspensions retain geometric track at all times.

On an independently suspended axle, track change occurs with normal suspension movement as shown in Figure 2.6. However, this condition is amplified due to tyre roll and *'tuck under'*.

As the vehicle rolls during cornering, the outside tyre will come under load due to the weight transfer effects. The inside tyre, in contrast, becomes unloaded but due to the geometry changes of the suspension, the track dimension can become biased left to right. This may be exaggerated if the vehicle hits a bump or pothole during this manoeuvre.

The phenomenon of tuck under of the tyres is due to the lateral compliance of the tyre sidewalls and the lateral acceleration generated during the cornering event. For handling purposes, very stiff sidewalls tend towards sharper response to steering inputs. However, the sidewall stiffness is a trade-off between handling characteristics and ride comfort (Curtis, 1983; Hinds, 1983; Pacejka, 2002).

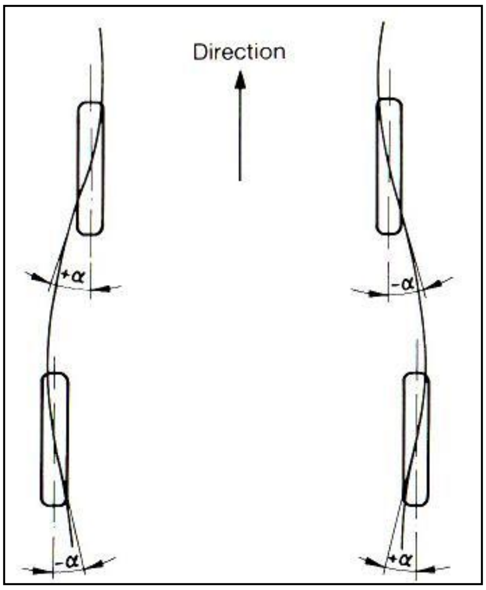



**Figure 2.6** – Dynamic Track Change (Reimpell et al, 1996)

The amount of tuck under the tyre will achieve is infinitely variable but it has been demonstrated that certain tyres with high aspect ratios can in some circumstances, shift the tread pattern up to half its section width (Loeb et al, 1990).

Figure 2.7 illustrates the effects of tyre tuck under. The vehicle on the left shows a vehicle with conventional springs during cornering. The vehicle on the right shows the same specification vehicle fitted with an active suspension system.

The active suspension system keeps the vehicle flat during cornering to counteract the weight transfer effects on the rolling vehicle to equalise the grip of the tyres on the same axle. As can be seen, this vehicle still exhibits some roll behaviour which is generated purely from tyre tuck under.

The tyres on the subject vehicle have a high aspect ratio (215/75 R16C) and may be subject to severe tuck under. Investigations will be carried out to determine the effect of tuck under and whether the tyres reaction to reduced vertical force due to weight transfer difference, affect this phenomenon.


**Figure 2.7** – Vehicle Roll in Combination with Tyre Tuck Under (Howard, 1987)

### 2.4.4 Steering

The feedback loop to the driver which provides handling confidence of a vehicle is the steering system. This mechanism provides all the drivers sensory inputs during transient manoeuvres. The steering geometry must be such that the amount of steering wheel angle required in a given situation is predictive and relatively linear. The steering system used on the subject vehicle is the standard rack and pinion layout with hydraulic power assistance.

The steering gear positioning in a vehicle influences the set up and ultimate development of the suspension for understeer / oversteer characteristics (Gillespie, 1992). Most front wheel drive (FWD) vehicles have the attachment of the steering gear behind the centreline of the wheels. This is purely to accommodate packaging of the steering gear with an *east-west* mounted engine. Rear steer, as this is known, has a tendency to roll oversteer during cornering which must be countered by the engineer during development (Gillespie et al, 1983).

Rear wheel drive vehicles generally have the steering attachment forward of the centreline of the front wheel and this geometry tends to roll understeer. However, the rear wheel drive Transit, because of its shared suspension geometry, maintains the FWD steering attachment behind the centreline of the wheel. It is possible that reduced body stiffness has reduced the weight transfer effects to a degree that the roll oversteer moments have become critical. This will be investigated analytically in Section 3.

Forces and moments acting on the steering system are shown in Figure 2.8. Front wheel drive vehicles impose an additional moment caused by the drive torque of the engine. It is the sum of the moments acting through the linkage which provide feedback to the driver and provide confidence (Whitehead, 1990).

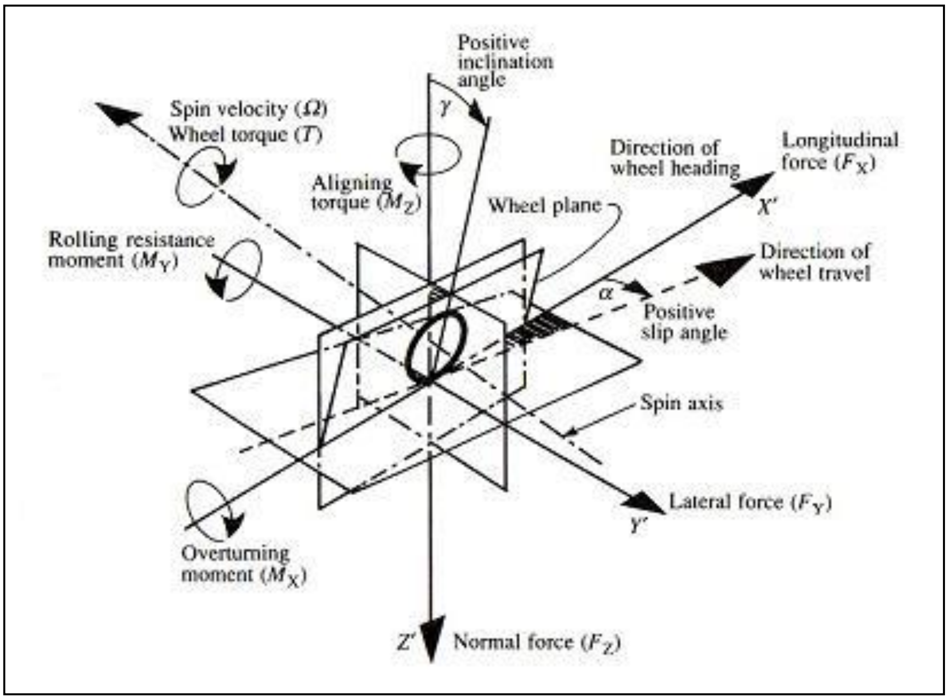



**Figure 2.8** – SAE Tyre Force and Moment Axes (Clark, 1981)

Investigations into steering analytical models have been conducted to analyse directional response and associated force and moments from inputs to the steering system. These can be used to determine steer angles and effects of turning behaviour of the vehicle. Precise influences of steering characteristics are then possible and will be investigated in Chapter 3 (Pitts et al, 1978)

Suspension behaviour can be divided into two subgroups:-

- Kinematics
- Compliance

### 2.4.5 Kinematics & Compliance (K&C)

Kinematics is suspension movement caused by roll, pitch, vertical and lateral movements of the vehicle body or road induced variations. As the potential for kinematic change has occurred due to body stiffness & weight transfer effects, kinematic behaviour of the suspension should be investigated.

Sophisticated kinematic rigs have been developed to establish accurate suspension movement. These rigs, initially developed for motor racing purposes, are now employed on road vehicles. The use of rigs to establish accurate kinematics is not widely used on road vehicles due to their cost. It is also difficult to separate compliance effects from pure kinematic movement of the suspension (Whitehead, 1996; Holdmann, 1998; Wright, 2001).

Analytical models are best suited for suspension kinematics analysis. Full details of the suspension, including bush details, can be incorporated into a model using a multi-body system. Kinematic properties of the suspension are represented in the model by functions defined by vertical wheel movement at the contact area.

A vehicle modelling tool such as ADAMS, Figure 2.9, can be used to specify the complete motion of a the vehicle as result of body movement and road induced variations (Holdmann et al, 1998). Analytical studies have been used to study derivatives of vehicles when physical testing would be expensive and time consuming. Analytical models also offer the advantage of being easily manipulated to allow the engineer to gauge the effects of different geometry relatively easily (Allen et al, 1987).

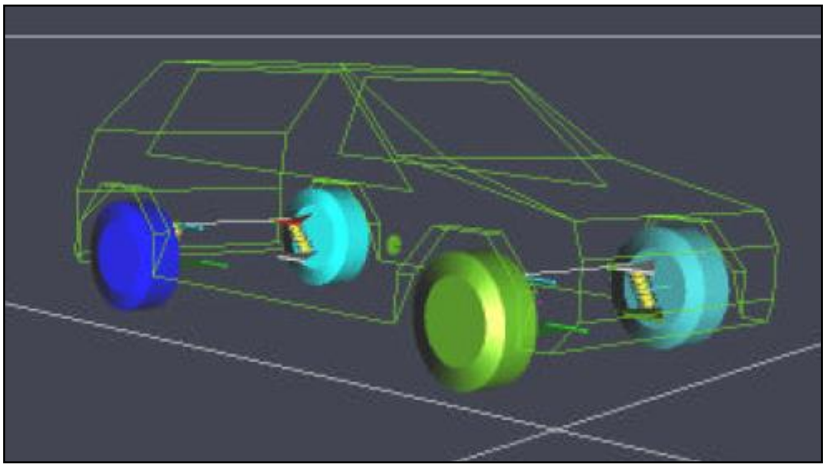


**Figure 2.9** – ADAMS Representation of Kinematics Model

*Compliance* is the science of elastic deformation of the suspension as a whole, including bushing assemblies, which are measured for degrees of freedom in the vertical and lateral modes as well as yaw.  Compliance sources allow the suspension to deflect forces inputted through the body or tyres.  These deflections cause camber, steer, vertical and lateral movement of the wheels due to compliance.  In the case of body stiffness influences, these effects must be quantified with and without RBB.

From the test results on the K&C rigs, it is possible for the vehicle designer to change the characteristics of the compliance by revising bushing rates and stiffnesses.  This then changes the handling characteristics of the whole vehicle.

Several studies have been undertaken to experimentally assess the road input influence on suspension design.  Vehicles are subjected to periodic, random and discrete inputs through the suspension in order to investigate the behaviour under such conditions (Kim et al, 1991).  It has been determined that periodic and random inputs generate random vibrations paralleling the input vibration.

Studies have been undertaken to investigate and ultimately predict loads into a suspension system.  These studies have been conducted analytically using three models (Giannopoulos et al; 1981):-

- Full vehicle model with 16 degrees of freedom capable of symmetric and no symmetric road input simulation
- Half vehicle model with 8 degrees of freedom capable of symmetrical road input simulation
- Suspension model with 2 degrees of freedom capable of symmetrical road input

Investigations into the effects of centre of gravity location have also been conducted to assess its affects to suspension compliance (Wong et al, 1988). These investigation carried out analytically, determined that centre of gravity location is not the predominant factor affecting dynamic compliance when compared with other parameters.

K&C analysis will be investigated analytically and physically (permitting adequate resources) to estimate the K&C variations with and without RBB. The results of this test may offer indications as to the possible causes and potential solutions for vehicle behaviour.

## 2.5 Experimental Methods

To establish a baseline for the current vehicle with RBB, it will be necessary to assess the vehicle using all of the previously mentioned theory and background. In order to assess the handling characteristics, a vehicle must be built and tested. This may be done by two methods:-

- Vehicle handling models (Analytical)
- Experimental vehicle handling (Physical)

### 2.5.1 Vehicle Handling Models (Analytical)

Journals detailing analytical development of vehicle models are varied and are readily accessible (Blundell, 1991 – 1999).

Studies into a simplified suspension model on the accuracy of calculated output from handling simulations have been undertaken. Experimental performance of a vehicle under lateral acceleration and roll has been compared with simulated 60° steering input (Blundell, 1991 – 1999).

Although variations between simulations and actual test data exist, these can be ascribed to non-uniform steering input or imperfection in track surface during data acquisition (Blundell, 1996).

Vehicle handling models vary from simple linear models to complex non-linear types. Modelling software tools are now currently available for developing these models, depending on the parameters required for analysis.

#### 2.5.1.1 Linear Models

A simple linear model is adequate for studying basic steering and handling responses. Non-linear tyre forces and suspension geometry effects are ignored in this approach. Although small steering inputs are available, only small amounts of lateral acceleration gain can be generated, making their use limited (Willumeit et al, 1992).

Linear handling models assume the following parameters (Dixon, 1991):-

- Small steering input
- Linear tyre behaviour (slip and camber angles)
- Lateral tyre force is insensitive to load changes
- Constant vehicle speed
- Smooth road surface
- Level road surface

Motions of vehicle on smooth roads and responses to wheel inputs has been studied with a degree of accuracy with lateral accelerations of 0.3g (Segel, 1956) but will not be considered in this thesis due to the non-linearity of the RBB effects on body stiffness.

#### 2.5.1.2 Non-Linear Models

The model that will be used to analyse RBB effects will be the non-linear model. Non-linear modelling includes the elements of the vehicle such as springs, dampers and tyres that are not linear. Non-linear models operating below 0.3g have to have behaviour patterns as linear models but about above 0.3g, and due to non-linear system inclusions, can be made to simulate real world vehicle conditions, subject to time history outputs. Non-linear models include the following parameters:-

- Tyre forces and moments
- Bump stop forces
- Rebound restriction forces
- Suspension kinematics
- Steering parameters
- Spring and anti-roll bar stiffness
- Damper parameters

Advanced vehicle dynamics simulation and correlation of vehicle design practicality has been discussed in numerous papers (Segel, 1956; Segel, 1993; Crolla, 1995; Butz et al, 2005). These investigations recognised computer software capable of performing non-linear analysis. These include:-

- Numerical based multi-body simulation packages such as ADAMS and veDYNA
- Algebraic multi-body simulation packages
- Purpose designed simulation codes
- Simulation toolkits such as MATLAB

Up until the 1930's, vehicle handling behaviour was not completely understood due to the lack of understanding of tyre deformity during turning events. From this period until the early 1950's, steady state turning behaviour was understood and factored into the governing factors of steady state handling. At this time, measurements were taken of transient handling behaviour and the equation of motion was developed. From this date, theoretical prediction into the direction response of moving vehicles was possible (Segel, 1993; Crolla, 1995).

These predictions achieved:-

- Linear mathematical modelling development with front wheel steering and validation
- Roll, yaw and side slip demonstration and their necessity to predict directional response to front wheel steering displacement which could be crucial in the yaw overshoot subjective effects of RBB deletion
- Understeer gradient expression developed analytically
- Quasi-static tyre demonstration validated
- Multi-body computer software for non-linear models developed and validated

Sensitivity analysis has been conduction to enhance computer simulation models by predicting an input parameter change required to cause a specific change in output variables. These results are given as yaw rate response and sensitivity as well as lateral acceleration when applied to a 13 degree of freedom model (Tandy et al, 1992).

Numerical models for steady state analysis to determine stability factors, side force coefficients and manoeuvring time for directional control have now been developed which include an unlimited manoeuvrable tyre model. These have been developed for linear and non-linear model (Allen, 1987) and will be used in the RBB deletion study.

The complexity of non-linear models has led to motion equations being prone to error. Multi-body system codes were developed to counteract these errors and allow automatic generation of motion equations for all mechanical constrained and dynamic systems. Dynamic responses of the vehicle are represented as time history outputs from the motion equation, provided mass and inertia data is identified from the following:-

- Rigid body systems
- Joints
- Springs
- Dampers
- Actuators

Multi-body codes such as ADAMS and veDYNA are widely used to generate and solve motion equations for models with hundred of degrees of freedom. These have found popular use in the study of vehicle dynamics (Rai et al, 1982; Antoun et al, 1986; McConville et al, 1984; Bartels et al, 1986; Hackert et al; 1986).

Assessment of multi-body codes and determination of the capabilities of each has not led to a definitive conclusion. Each study carried out independently cites advantages and disadvantages of each without a clear indication as to the best approach for analytical study of vehicle dynamics (Sharp, 1991; Kortum, 1993; Sharp, 1994; Sharp, 1998).

Multi-body dynamic codes have advantages in applications of non-linear vehicle model simulations:-

- Motion equations generated and solved automatically
- Static equilibrium condition of a model can be determined
- Rigid body connection properties easily identified
- Vehicle suspensions can be analysed and bushings optimised for ride and handling trade-off

The multi-body dynamic system used in the thesis will be ADAMS due to its widespread and common use throughout the automotive industry and the availability of an ADAMS model of the subject vehicle. This will be manipulated to understand the effects of the dynamic behaviour of the vehicle with and without RBB.

### 2.5.2 Experimental Vehicle Handling (Physical)

As stated in Section 2.3, experimental vehicle handling relies upon experienced vehicle assessors making subjective evaluations pertaining to the vehicles handling characteristics. Manufacturers continue to use expert driver assessors as computer simulations and mathematical modelling assessments still can not predict accurate *feel* assessments of a vehicle (Crolla et al, 1998).

Despite the advances in modelling and the correlations proven between analytical and physical vehicle development, all vehicles are driven subjectively prior to sale in the market place. Motoring journals such as Autocar and Auto Express publish subjective evaluations of most vehicles before they go on sale and so it is imperative that manufacturers have good subjective ratings established for any vehicle prior to testing by outside influences and use by their customers (Crolla et al, 1997; Crolla et al, 2001).

To that end, a subject vehicle will be used to verify physical effects of RBB deletion (K&C rig testing, jury evaluation etc), and to complement analytical testing

Most subjective vehicle evaluations rely on a scale of 1 to 10 to represent scales of performance, Figure 2.10.

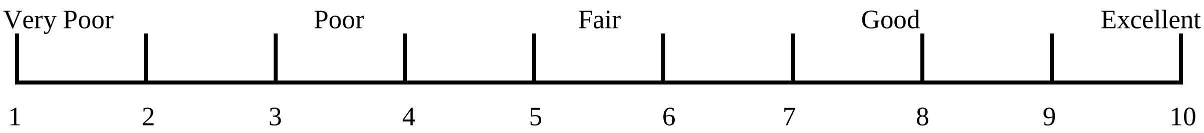



**Figure 2.10** – Subjective Rating Scale

Rating scales have been used successfully for ride comfort evaluations, dynamic assessments and drivability assessments by most major vehicle manufacturers and variations of the scale exist throughout the world (Crolla et al, 2001). The Ford motor Company R202 Vehicle Evaluation Rating standard will be used throughout this thesis with regard to vehicle evaluation and rating, Figure 2.11.

**VEHICLE RATING SYSTEM**

| | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | UNACCEPTABLE | | | | BORDER LINE | ACCEPTABLE | | | | |
| RATING INDEX | **1** | **2** | **3** | **4** | **5** | **6** | **7** | **8** | **9** | **10** |
| EVALUATION OF VEHICLE & COMPONENT PERFORMANCE | PRODUCTION REJECT | | | | BORDER LINE | ACCEPTABLE | FAIR | GOOD | VERY GOOD | EXCELLENT |
| | POOR | | | CUSTOMER COMPLAINT | | | | | | |
| CONDITION NOTED BY | ALL CUSTOMERS | AVERAGE CUSTOMER | | | CRITICAL CUSTOMER | | | TRAINED OBSERVER | | NOT PER-CEPTIBLE |
| NOISE, VIBRATION, HARSHNESS, SHAKE, ETC. | NOT ACCEPTABLE | | | OBJECT-IONABLE | REQUIRES IMPROVE-MENT | MEDIUM | LIGHT | VERY LIGHT | TRACE | NOT NOTICE-ABLE |

**Figure 2.11** – Ford Motor Company Vehicle Evaluation Rating Scale (Ford Corporate Test Procedure, 2001)

## 2.6 Summary

As seen from the review of available literature, careful study is essential to determine the subjective and objective correlations between ratings and measurements for transient manoeuvres.

It is important to establish the tyre models for transient handling events as well as the vehicle handling characteristics such as neutral steer, understeer and oversteer.

Non-linear full vehicle models have been developed for the purpose of analysing transient handling response.  Four fundamental objectives to successfully develop a full vehicle model are:-

1. Front and rear suspension model establishment for vehicle handling simulation
2. Non-linear force and moment modelling at the occurrence at the road and tyre contact patch
3. Non-linear system modelling for springs, dampers, tyres, bump stops and rebound restrictors
4. Accurate modelling of the vehicle body as a flexible element to determine weight transfer coefficients

From the research conducted, it is clear that the transient handling element of vehicle dynamics must be considered due to the problem definition in Section 1. RBB deletion and its effect on lateral as well as longitudinal weight transfer will also be investigated analytically and physically (where possible).

Body hysteresis will also be investigated as very little research into this phenomenon has been conducted in the past in relation to perceived vehicle dynamics.

Investigations to be conducted from the literature search are:-

- Simple torsional rigidity test for body compliance
- Determine co-efficient of torsional rigidity vs total roll stiffness
- Determine hysteresis of current vehicle with and without RBB. Establish dynamic behaviour prediction
- Conduct CAE study of current vehicle body to determine stress concentrations and improvement areas
- Conduct jury trials of driver evaluation of body structure feel with differing hysteresis properties
- Apply control theory method to predict roll, yaw and lateral acceleration with 3 degree of freedom model
- Determine tyre tuck under influence. Determine best method of analysis for future work:-
    - Physically
    - Analytically
- Determine weight transfer effects of body without RBB and the influence of roll oversteer due to steering geometry (rear steer).
- Determine hysteresis results with and without RBB:-
    - Physically
    - Analytically

# 3. Vehicle Parameter Investigation

## 3.1 Introduction

The following chapter investigates influences on vehicle behaviour suggested in Chapter 2. The following vehicle parameters will be evaluated for their magnitude of influence over vehicle yaw response with and without an RBB being fitted:-

- Torsional body compliance
- Co-efficient of torsional rigidity vs total roll stiffness determination
- Tyre dynamics
- Steering geometry effects on weight transfer

The following analytical studies will also be carried out:-

- Stress concentration determination on BIW
- Force and moment change influence on directional response

In addition, analytical testing will be performed to quantify the effects of vehicle body response without driver and road influences.

## 3.2 Torsional Body Compliance

Torsional loading of a vehicle body produces lateral as well as longitudinal load transfer. To establish the load transfer effects on the subject vehicle, a simple test was devised that induced torsional loads whilst vehicle corner weights were recorded. This test was done at the laden condition, with and without the RBB fitted.

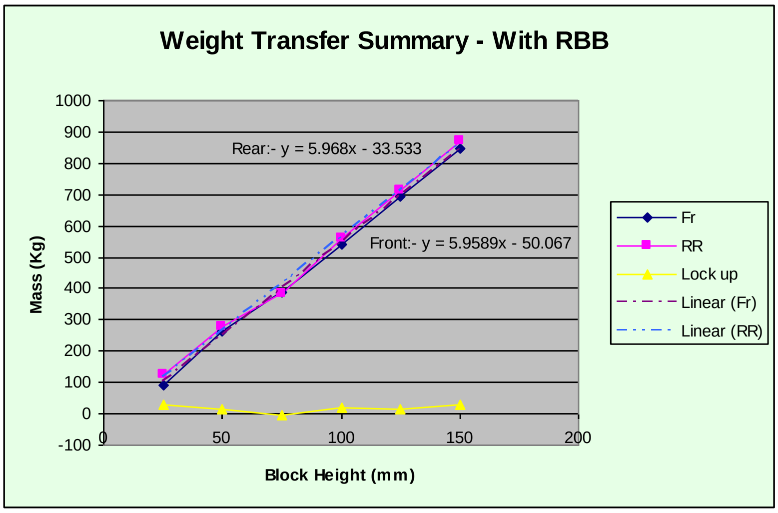
A vehicle was driven onto a weighbridge and progressively *chocked* at each diagonal wheel in increments of 25mm up to 150mm, see Figure 3.1. The individual corner weights were measured at each increment. The results of this test are shown in Table 3.1.

| Wheel Height (mm) | Axle Mass (kg) | With RBB | | | Without RBB | | |
|---|---|---|---|---|---|---|---|
| | | **Left** | **Right** | **Total** | **Left** | **Right** | **Total** |
| 0 | Front | 718 | 758 | 1476 | 721 | 754 | 1475 |
| | Rear | 1021 | 1003 | 2024 | 1023 | 999 | 2022 |
| 25 | Front | 675 | 807 | 1482 | 654 | 822 | 1476 |
| | Rear | 1080 | 939 | 2019 | 1085 | 936 | 2021 |
| 50 | Front | 586 | 890 | 1476 | 596 | 882 | 1478 |
| | Rear | 1161 | 864 | 2025 | 1152 | 869 | 2021 |
| 75 | Front | 526 | 955 | 1481 | 502 | 971 | 1473 |
| | Rear | 1212 | 809 | 2021 | 1234 | 793 | 2027 |
| 100 | Front | 447 | 1030 | 1477 | 455 | 1027 | 1482 |
| | Rear | 1302 | 723 | 2025 | 1315 | 704 | 2019 |
| 125 | Front | 370 | 1104 | 1474 | 391 | 1085 | 1476 |
| | Rear | 1379 | 650 | 2029 | 1372 | 652 | 2024 |
| 150 | Front | 295 | 1181 | 1476 | 334 | 1154 | 1488 |
| | Rear | 1460 | 569 | 2029 | 1424 | 590 | 2014 |

**Table 3.1** – Weight Transfer Effects of Torsional Loading

**Figure 3.1** – Torsional Load Inducement by Wheel *Chocking*

When plotted, the load distribution is show to be roughly linear, Figure 3.2.



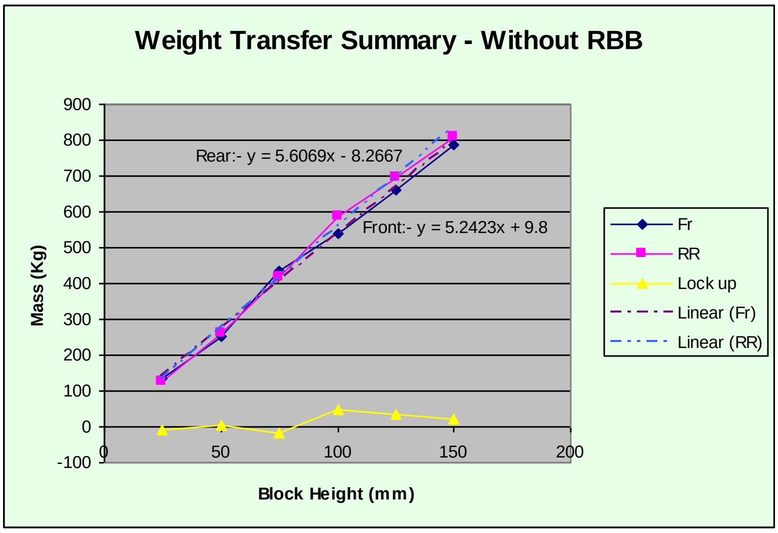




**Figure 3.2** – Graphical Representation of Weight Distribution Test

Although the weight transfer is different between the two vehicle states, the percentage difference is not considered large enough to be a contributing factor, Table 3.2. The differences discovered between the two tests are consistent with friction build up variation in the suspension members and can therefore be discounted.

| Wheel Height (mm) | Axle Mass (kg) | With RBB | Without RBB | Percentage Difference |
|---|---|---|---|---|
| | | **Total** | **Total** | **%** |
| 0 | Front | 1476 | 1475 | **0.001** |
| | Rear | 2024 | 2022 | **0.001** |
| 25 | Front | 1482 | 1476 | **0.004** |
| | Rear | 2019 | 2021 | **-0.001** |
| 50 | Front | 1476 | 1478 | **-0.001** |
| | Rear | 2025 | 2021 | **0.002** |
| 75 | Front | 1481 | 1473 | **0.005** |
| | Rear | 2021 | 2027 | **-0.003** |
| 100 | Front | 1477 | 1482 | **-0.003** |
| | Rear | 2025 | 2019 | **0.003** |
| 125 | Front | 1474 | 1476 | **-0.001** |
| | Rear | 2029 | 2024 | **0.002** |
| 150 | Front | 1476 | 1488 | **-0.008** |
| | Rear | 2029 | 2014 | **0.007** |

**Table 3.2** – Percentage Difference of Load Transfer Test

When the two graphs are overlaid, the percentage difference calculations can be visualised.  Although the weight transfer effects are different, the overall trends are essentially common, Figure 3.3.

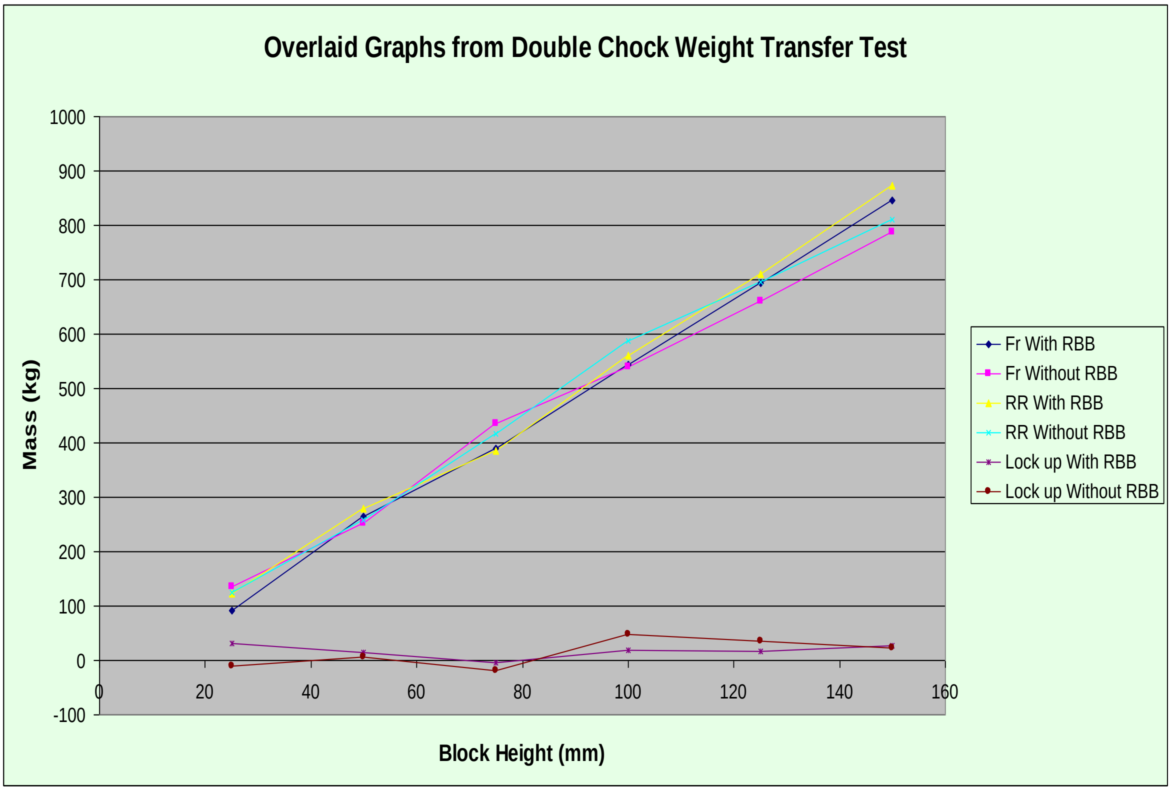



**Figure 3.3** – Overlaid Double Chock Test Graph

This data can be compared directly with the theoretical data from ADAMS.  As there is no body information or suspension friction allowance within ADAMS, the theoretical weight transfer effects can be determined and compared against the physical data, Table 3.3.  It should be noted however that the ADAMS model used in this study simulated corner weights without body stiffness interactions.

| Wheel Height (mm) | Axle Mass (kg) | Left | Right | Total |
|---|---|---|---|---|
| 0 | Front | 749 | 749 | 1498 |
| | Rear | 1005 | 1005 | 2010 |
| 25 | Front | 658 | 841 | 1498 |
| | Rear | 1099 | 911 | 2010 |
| 50 | Front | 577 | 922 | 1498 |
| | Rear | 1184 | 825 | 2010 |
| 75 | Front | 495 | 1004 | 1498 |
| | Rear | 1272 | 738 | 2010 |
| 100 | Front | 412 | 1086 | 1498 |
| | Rear | 1359 | 651 | 2010 |
| 125 | Front | 328 | 1169 | 1498 |
| | Rear | 1447 | 563 | 2010 |
| 150 | Front | 254 | 1242 | 1498 |
| | Rear | 1523 | 487 | 2010 |

**Table 3.3** – ADAMS Theoretical Weight Transfer Effects

These results have the following graphical representation, Figure 3.4.

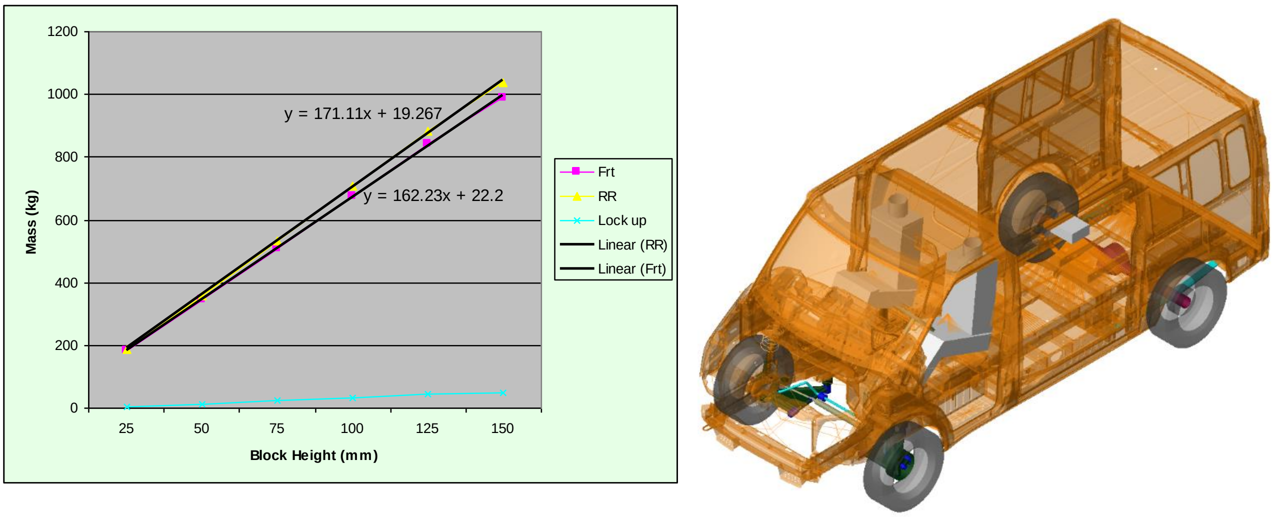



**Figure 3.4** – Graphical Representation of Theoretical Weight Distribution Test in ADAMS

The weight transfer effects in ADAMS, correlate with the weighbridge test inasmuch as the percentage error is not enough to quantify the effects of RBB deletion on global torsional stiffness.

Speculation as to why the vehicle did not demonstrate any significant weight transfer change centred around the test method itself. As opposite wheels were chocked at the same time, the measurements could be centred around body bending about a *beam* rather than measuring pure twist, Figure 3.5. The test was therefore re-run chocking one wheel only, Figure 3.6.

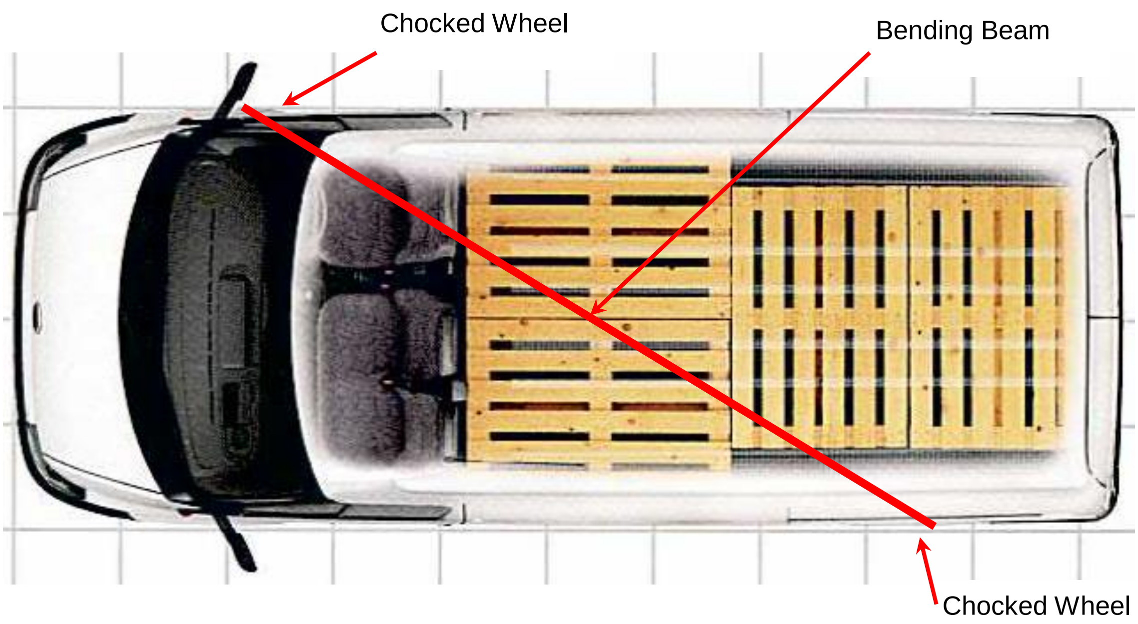



**Figure 3.5** – Vehicle Bending Mode

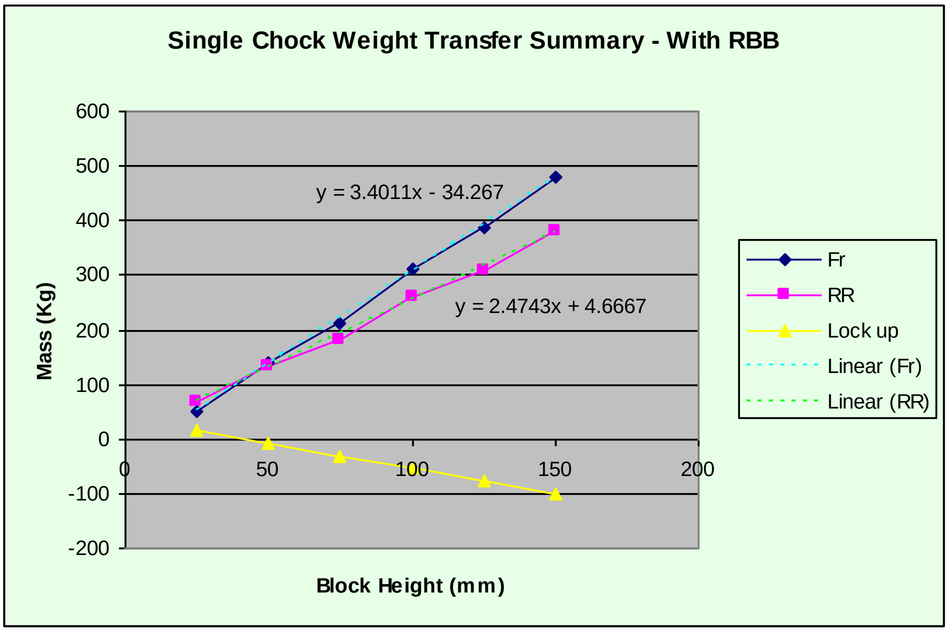
The results of the chocking test on one rear wheel are shown in Table 3.4.

| Wheel Height (mm) | Axle Mass (kg) | With RBB | | | Without RBB | | |
|---|---|---|---|---|---|---|---|
| | | Left | Right | Total | Left | Right | Total |
| 0 | Front | 718 | 758 | 1476 | 718 | 758 | 1476 |
| | Rear | 1021 | 1003 | 2024 | 1022 | 1002 | 2024 |
| 25 | Front | 693 | 785 | 1478 | 685 | 792 | 1477 |
| | Rear | 1054 | 968 | 2022 | 1050 | 972 | 2022 |
| 50 | Front | 650 | 830 | 1480 | 640 | 840 | 1480 |
| | Rear | 1085 | 935 | 2020 | 1083 | 936 | 2019 |
| 75 | Front | 616 | 867 | 1483 | 607 | 875 | 1482 |
| | Rear | 1108 | 910 | 2018 | 1107 | 910 | 2017 |
| 100 | Front | 567 | 918 | 1485 | 555 | 929 | 1484 |
| | Rear | 1147 | 870 | 2017 | 1147 | 870 | 2017 |
| 125 | Front | 530 | 957 | 1487 | 520 | 965 | 1485 |
| | Rear | 1171 | 844 | 2015 | 1170 | 847 | 2017 |
| 150 | Front | 485 | 1004 | 1489 | 487 | 998 | 1485 |
| | Rear | 1205 | 808 | 2013 | 1193 | 825 | 2018 |

**Table 3.4** – Single Chocked Wheel Weight Transfer Effects of Torsional Loading

**Figure 3.6** – Torsional Load Inducement by Single Wheel Chocking

When plotted graphically, the load distribution is shown to be roughly linear, Figure 3.7.



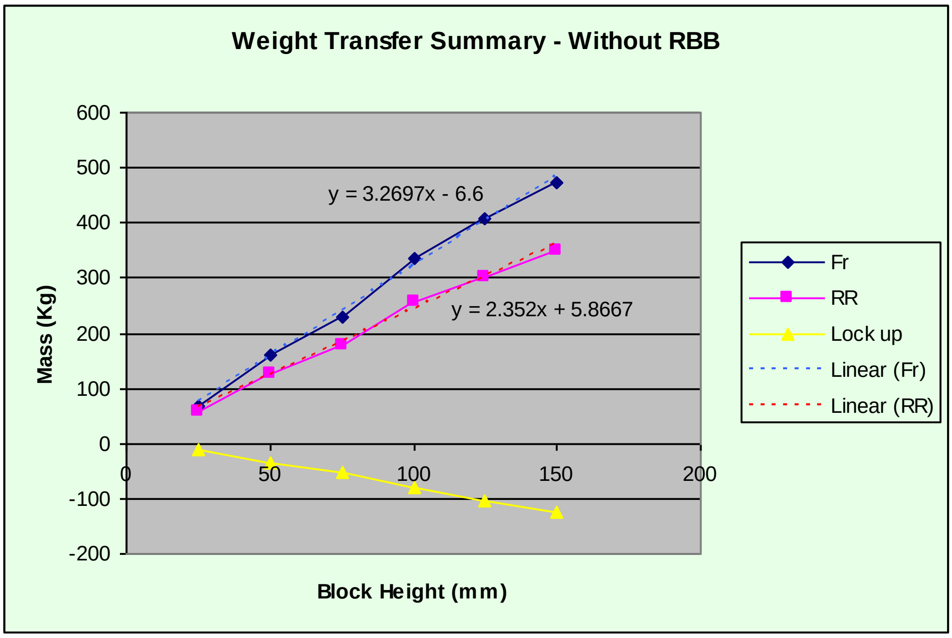




**Figure 3.7** – Graphical Representation of Single Chock Weight Distribution Test

The percentage differences between the two vehicle conditions are similar to the double chocked wheel test, Table 3.5, however, the twisting motion of this test requires further investigation as this is more likely to highlight the torsional response of the RBB and its contribution to global stiffness.

| Wheel Height (mm) | Axle Mass (kg) | With RBB | Without RBB | Percentage Difference |
|---|---|---|---|---|
| | | **Total** | **Total** | **%** |
| 0 | Front | 1476 | 1476 | **0.000** |
| | Rear | 2024 | 2024 | **0.000** |
| 25 | Front | 1478 | 1477 | **0.001** |
| | Rear | 2022 | 2022 | **0.000** |
| 50 | Front | 1480 | 1480 | **0.000** |
| | Rear | 2020 | 2019 | **0.001** |
| 75 | Front | 1483 | 1483 | **0.000** |
| | Rear | 2018 | 2017 | **0.001** |
| 100 | Front | 1485 | 1484 | **0.001** |
| | Rear | 2017 | 2017 | **0.000** |
| 125 | Front | 1487 | 1485 | **0.002** |
| | Rear | 2015 | 2017 | **-0.001** |
| 150 | Front | 1489 | 1485 | **0.003** |
| | Rear | 2013 | 2018 | **-0.002** |

**Table 3.5** – Percentage Difference of Single Chock Load Transfer Test

When the two graphs are overlaid, the percentage difference calculations can be visualised. Although the weight transfer effects are similar comparing front and rear results, the overall weight transfer differences (rear to front) are clearly different, Figure 3.8.

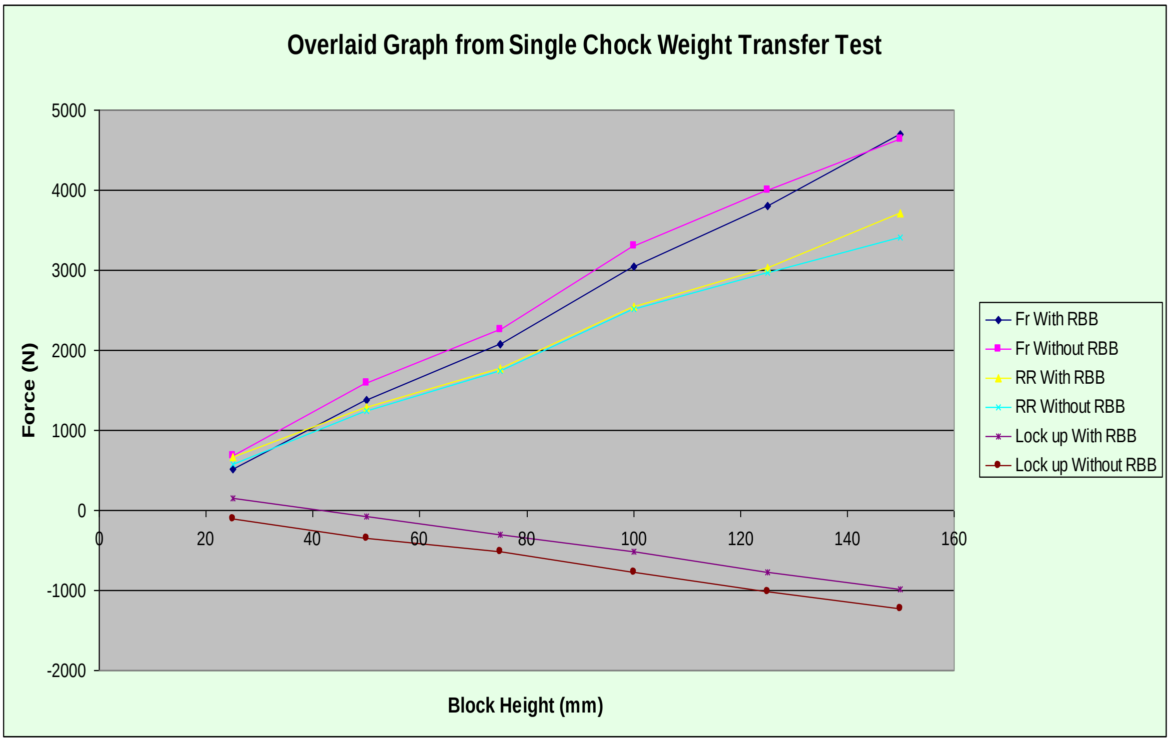



**Figure 3.8** – Overlaid Single Chock Test Graph Converted to Force vs Displacement

By integrating both front and rear axle curves, it is possible to determine the work done by the front and rear wheels with and without the RBB. Although this number itself does not determine the route cause of the issue, it is an indicator as to the amount of work being done by the RBB that contributes to total weight transfer.

Table 3.6 details the work contribution of the RBB.

| WORK DONE (J) | | | | | |
|---|---|---|---|---|---|
| FRONT | | | REAR | | |
| DISPL mm | WITH RBB | WITHOUT RBB | DISPL mm | WITH RBB | WITHOUT RBB |
| 25 | 10791 | 11404 | 25 | 7848 | 8461 |
| 25 | 30288 | 31147 | 25 | 21582 | 23054 |
| 25 | 51257 | 52484 | 25 | 37155 | 38995 |
| 25 | 72839 | 74188 | 25 | 52974 | 54446 |
| 25 | 93440 | 90988 | 25 | 67689 | 65604 |
| TOT WD | 258616 | 260210 | TOT WD | 187248 | 190559 |
| CONTRIBUTION | 1594 | | CONRIBUTION | 3311 | |

**Table 3.6** – RBB Contribution to Weight Transfer

These results clearly show more contribution at the rear, although the front contribution has the same order of magnitude. The conclusion from this analysis is that the RBB deletion does corrupt the front suspension integrity and the reasons for this must be determined.

## 3.3 Body Torsional Contribution to Total Roll Stiffness

In order to transmit rolling torque during cornering, the body must be adequately stiff to allow the suspension to transfer weight laterally *and* longitudinally. The body stiffness contribution to total roll stiffness varies by manufacturer but can be estimated for commercial vehicles by determining the body stiffness and suspension stiffness requirements.

As has been stated in Section 2.2.1., studies into racing vehicles has suggested that relationships between suspension stiffness and global stiffness are potentially more significant than front to rear suspension stiffness alone (Kang et al, 1991; Deakin et al, 2000). A racing vehicle should have body stiffness of between 1/2 - 2/3rds of the total roll stiffness.

The subject vehicle has a body torsional rigidity value of 794 kNm / Radian. As the vehicle has acceptable handling with the RBB attached, it must be assumed that this level of body stiffness is adequate for transmission of weight transfer torque via the suspension. The roll stiffness contribution of the suspension to global stiffness is in the region of 80kNm / Radian or approximately 10% of global stiffness. This data is key as suspension stiffness will be constant to allow body stiffness variation to be calculated accurately.

Taking this contribution as the minimum acceptance criteria, it suggests that the RBB deletion has reduced the body stiffness by an amount sufficient to reduce the weight transfer. The single chocked weight transfer test also indicates that this hypothesis is correct, Figure 3.8.

Analytical studies into the effects of the RBB in a body-in-prime, however, have concluded that the RBB has minimal body stiffness contribution, Figure 3.9 (Harvey, 2004).

These results show that not only does the RBB not influence torsional stiffness but that global modes are also unaffected by its presence. In fact, the analytical model shows that torsional stiffness and the global torsion mode are ***increased***.

| V347 | Units | SWB LR Pre<PA> with rr bumper | SWB LR Pre<PA> without rr bumper | Percentage difference |
|---|---|---|---|---|
| Mass | Kg | 480.8 | 474.0 | -1.4% |
| Torsional Stiffness | kNm/rad | 794 | 796 | 0.2% |
| GLOBAL MODES | | | | |
| Global torsion | Hz | 21.53 | 21.73 | 0.9% |
| Vertical bending | Hz | 38.81 | 38.83 | 0.0% |
| Lateral bending | Hz | 44.09 | 44.09 | 0% |

**Figure 3.9** – RBB Body Stiffness Contribution

It is known however that the Natsran model used in the analysis has a ± 0.5% rounding error built into the software. Also, when considering that the resonant frequency of a body is proportional to the square root of the product (stiffness / mass), it follows that a reduction in mass (via RBB deletion) would increase the global torsion mode. Ie.

$$f(n) = K \bullet \sqrt{\frac{stiffness}{mass}}$$

Where $K$ is a constant - **Equation 1**

It should be noted however that the FEA approach to body stiffness contribution is analysed in the same test set-up as physical testing. This is to establish correlation to ensure reliable analytical results.

When vehicle body torsional stiffness is measured, the body is set up on a rig with attachment points location at the front and rear shock absorber positions, Figure 3.10. The front axle sits on a pivot arm than can induce twist forces into the body thus allowing torsional loads to be measured via strain gauges placed throughout the body, Figure 3.11.

In the case of RBB deletion, the twist is likely to be more severe at the rear of the vehicle to account for the differences in stiffness with and without the RBB.

However, as can be seen, the vehicle is located at four points; points 19 (front strut attachment) and points 4 (rear shock absorber attachments). As the pivot arm twists, moments through the vehicle are reduced the further from the fulcrum they are placed.

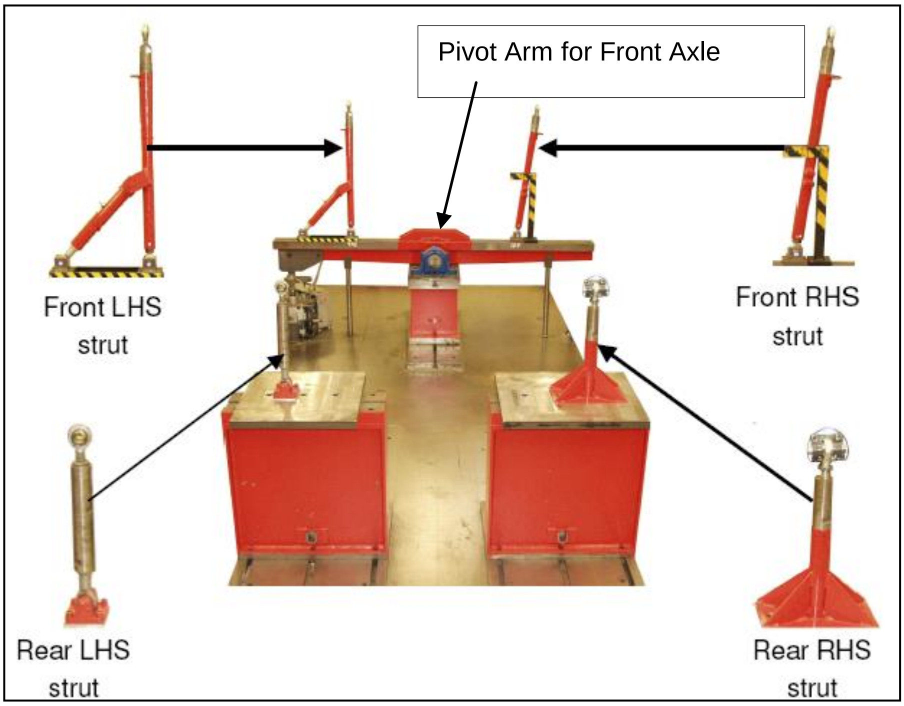



**Figure 3.10** – Torsional Rigidity Test Rig (Harvey, 2004)

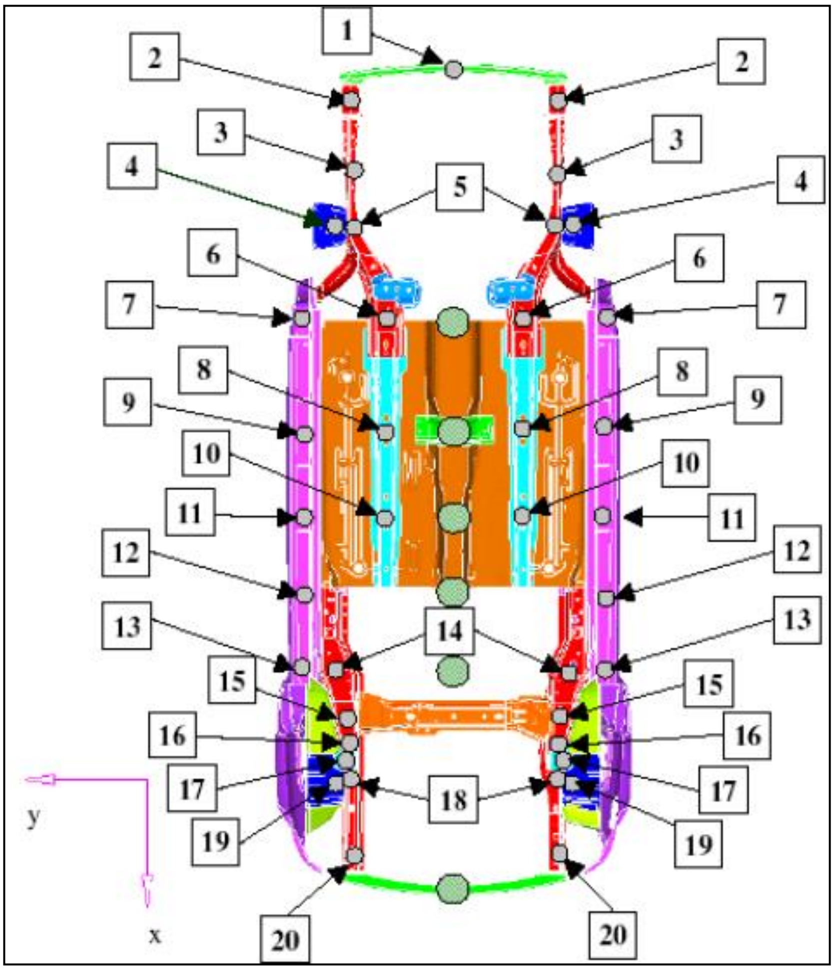



**Figure 3.11** – Strain Gauge Placements (Harvey, 2004)

Points 2 & 3 are located furthest away from the fulcrum and as such any strain loads, although measured as a variance independently, are likely to be discounted due to static friction when compared with previous results from other tests.

## 3.4 Tyre Dynamics

The dynamics of a tyre during transient manoeuvres are difficult to measure on vehicle due to the rolling nature of the components. The effects of tyre tuck under are especially difficult to assess as these happen rapidly in the transient state.

The rate of tyre tuck under will vary on many conditions:-

- Rate of steer
- Vehicle speed
- Ambient temperature
- Track temperature
- Slip Angle
- Pneumatic trail
- Tyre pressure
- Tyre construction
- Tyre compound
- Tyre Manufacturer

Of these conditions, ambient temperature and track temperature are key elements that can be varied to allow a design of experiments (DOE) to determine the influence of tyre tuck under.

For tuck under to be shown as an influence, it must be demonstrated that the deletion of the RBB has changed the rate at which the tyres friction properties are *saturated* during transient manoeuvres. Therefore, changing the glassation point of the tyre by varying the temperature, in theory, should demonstrate this phenomenon; as the temperature will not only change the slip characteristics but also the stiffness of the tyres sidewalls (Triton Technologies, 2008).

As the yaw overshoot phenomenon is present under all driving conditions including seasonal variations, temperature can be discounted as a major factor; as colder temperatures would reduce the tendency of tuck under as the sidewalls would remain stiffer in these conditions.

Also, commercial vehicles are fitted as standard with a variety of different brands of tyre. These can be chosen by the customer depending on preference as well as by market but the phenomenon exists in all cases and all variations – including markets where winter tyres are mandatory.

For these reasons, tyre tuck under can be discounted as a contributing factor.

## 3.5 Steering Geometry Effects on Weight Transfer

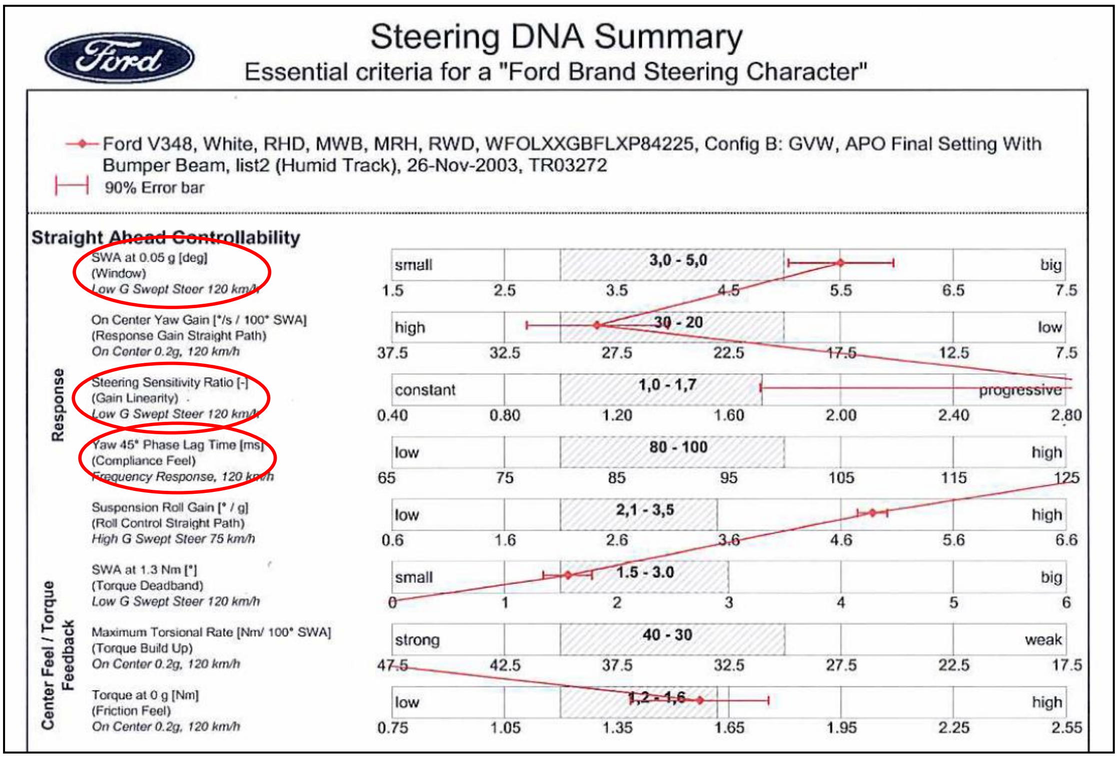
As body stiffness contribution of the RBB can be established based on the chocking test results, roll oversteer effects due to reduced body stiffness may also be established, as the steering characteristics of a vehicle with and without RBB can be measured objectively.

Steering DNA studies have shown that the RBB has a measurable effect on straight ahead controllability, especially in the low G swept steer and frequency response at 120km/h, Figure 3.12.

Steering DNA characterisation is performed using a steering robot in order to remove driver variability in the test results. Vehicle responses can therefore be shown to correlate with the driver reports of increased yaw overshoot at the same speed. It has been shown that global body stiffness may contribute to a steering loss function, however, alternative causes must also be evaluated.

Traditionally, assuming the same vehicle parameters, seven influences on steering function loss are:-

- Tyre performance and characteristics
- Load transfer
- Mass
- Body stiffness
- Prevailing road conditions
- Driver inputs (Steer, brake & acceleration)
- Wind



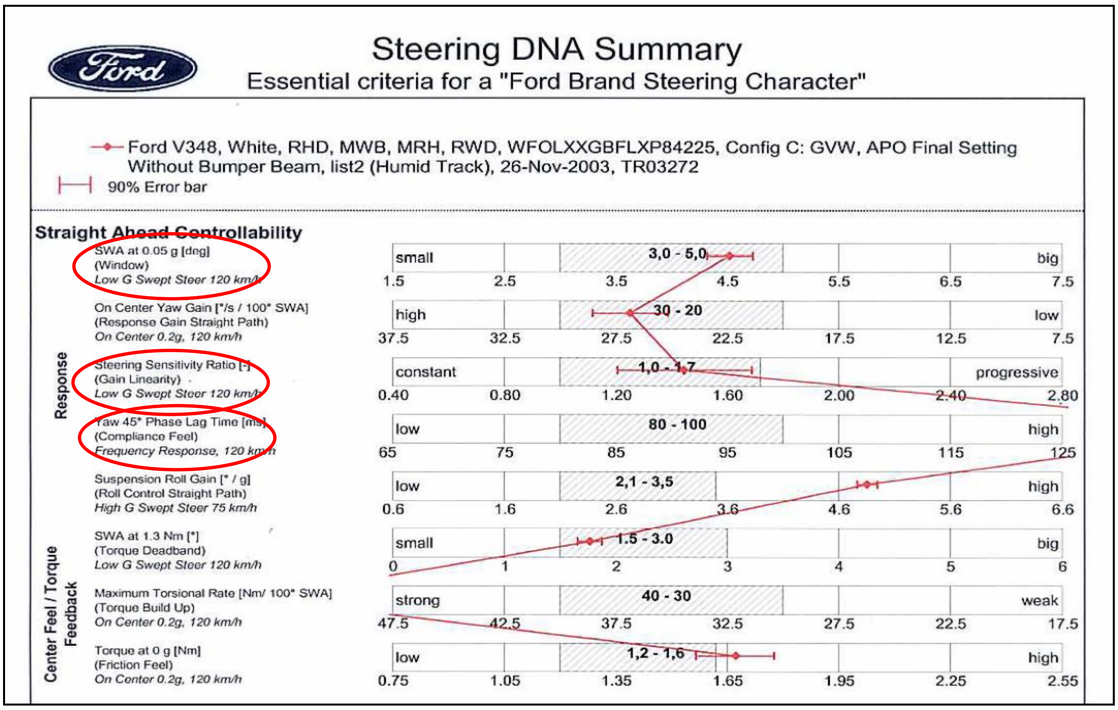




**Figure 3.12** – Steering DNA Characteristics With and Without RBB

### 3.5.1 Tyre Performance and Characteristics

Although tyre slip angles vary from manufacturer to manufacturer, the RBB yaw overshoot effect is evident on so many vehicles, passenger cars as well as commercial vehicles, that tyre differences can effectively be ruled out. It is also not possible to solve the yaw overshoot issue purely by developing the tyres for two reasons:-

1. Commercial issues with all tyre suppliers prevent sourcing the entire vehicle range of tyres to one supplier. Developing all manufacturers to have the same tyre characteristics is undesirable.
2. Customers, particularly fleet customers, can use any tyre on the market for replacement items and expect the vehicle handling behaviour to remain unchanged. Developing one tyre for the vehicle would not meet this customer need.

### 3.5.2 Load Transfer

As has been demonstrated in section 3.2., load transfer differences do exist during dynamic events that cause twisting of the vehicle body. As in any event that causes load transfer, the understeer gradients can be determined. The understeer gradient for any vehicle can be calculated by the formula in Equation 2:-

$$K = \frac{W_f}{C_{af}} - \frac{W_r}{C_{ar}}$$ - **Equation 2**

Where:-

$W_f$ = Static weight of front axle

$W_r$ = Static weight of rear axle

$C_{af}$ = Tyre force of front axle

$C_{ar}$ = Tyre force of rear axle

This equation, whilst taking tyre force variation and front a rear axle weights into account from a standard bicycle model, does not allow for body stiffness variations. In fact, most vehicle handling equations are derived assuming a body that is infinitely stiff such that all weight transfer effects are transposed directly to the suspension kinematics and compliances. This is clearly not the case in this subject.

The understeer gradients therefore need to be derived empirically during drive events with instrumented vehicles.

The vehicle with the RBB fitted has an understeer gradient (USG) of 1.52 °/g, Figure 3.13

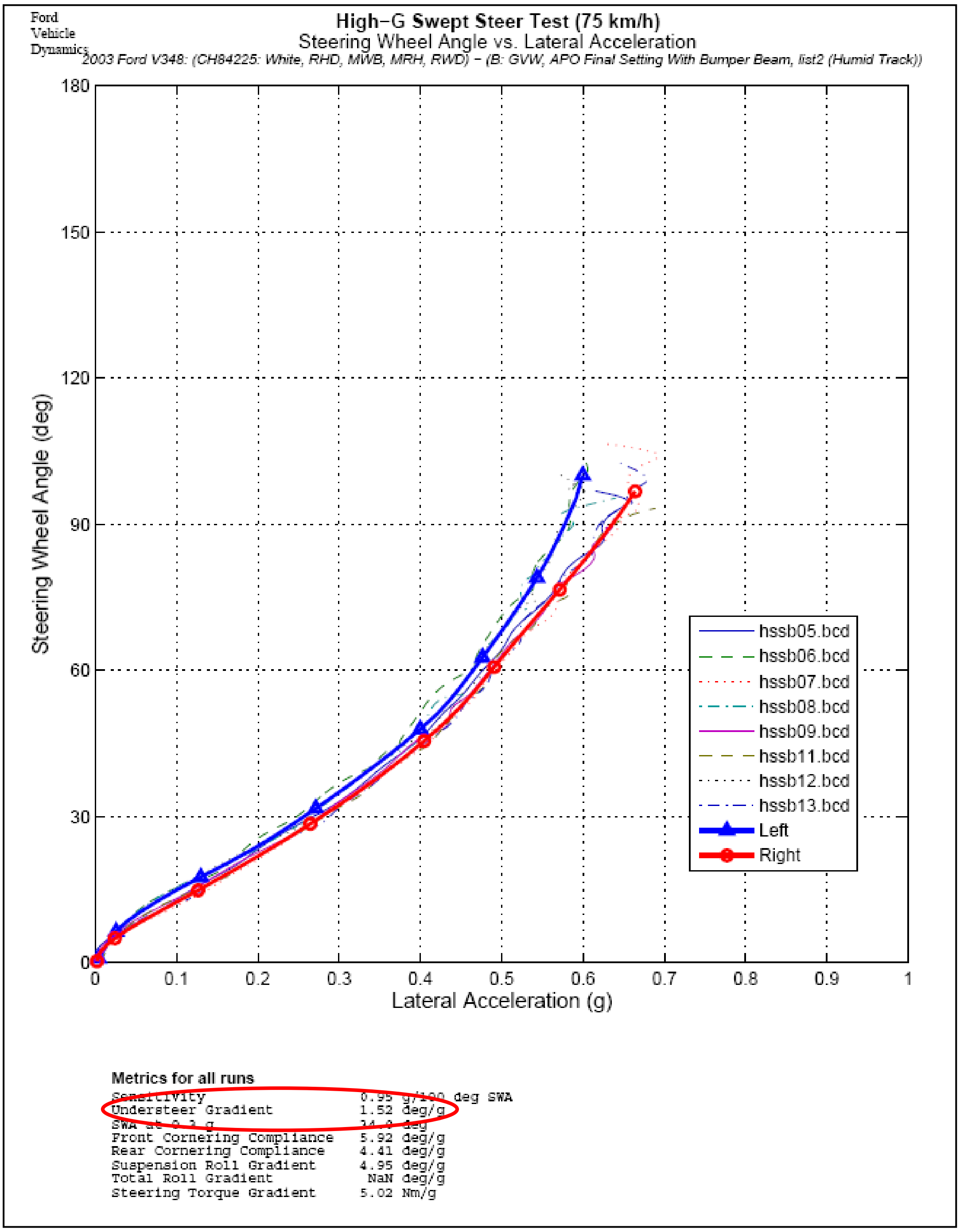



**Figure 3.13** – Understeer Gradient with RBB Fitted

The vehicle without the RBB fitted has an understeer gradient of 1.69 °/g, Figure 3.14

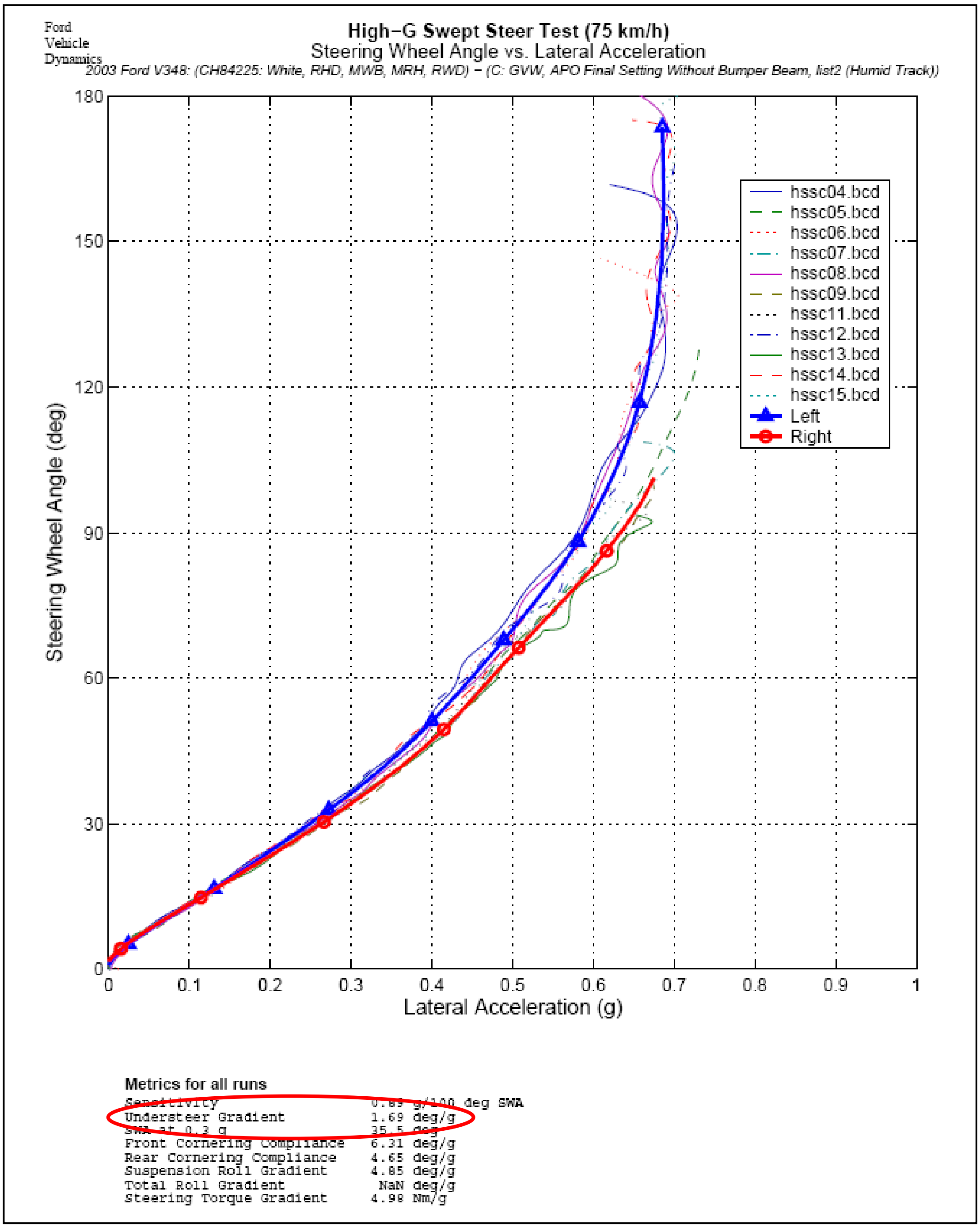



**Figure 3.14** – Understeer Gradient without RBB Fitted

Although the *USG* changes depending on load conditions among other variables, the value of the USG can give an indication as to the behaviour of a vehicle (assuming consistent test conditions).

As $K > 0$, the vehicle will tend towards understeer as the tyre slip angle will increase to maintain the vehicle path around a given radius.

If $K = 0$, the vehicle will maintain it course around a given radius as the slip angle is not increased. This is known as *neutral steer* or the *ideal case*.

If $K < 0$, the vehicle will tend towards oversteer as the tyre slip angle must be reduced to maintain a path around a given radius.

As can be seen, the vehicle tested without RBB tends to understeer more; with the USG figure increasing by 10%. In the past, this variation has been explained by variations in test conditions, suspension hysteresis and measurement anomalies. However, when these results are taken into account together with the weight transfer effect differences in Table 3.6, it suggests that understeer, and therefore the total grip available to the front tyres, is reduced by the deletion of the RBB.

Tyre grip is approximately linear to load until the tyres reach their saturation point as shown in Figure 2.4. After this point, the tyres will reduce in grip as the vehicle loses traction either in understeer or oversteer.

As the drive event that displays yaw overshoot variation can be measured during moderate steering events, it can be determined that tyre grip is being affected during the linear stage and that any increased understeer is not due to tyre saturation beyond Point A on Figure 2.4.

This is important as the determination of weight transfer between the front tyres can be more accurately studied in the linear range.

### 3.5.3 Mass

As the mass of the vehicle is changed when the RBB is deleted, it may be argued that the difference in mass has an effect in transient handling, especially when it is remembered that the RBB is located at the extreme rear of the vehicle. However, effects of mass were discounted as the RBB itself weighs 6kg which is only 0.17% of a fully laden vehicle (3500kg). A test was performed with masses in place of the RBB equal to the RBB itself but unconnected, Figure 3.15. The vehicle performance was unchanged in this configuration.


**Figure 3.15** – Vehicle with Mass Points at RBB Mounting Locations

### 3.5.4 Body Stiffness

As has been demonstrated in section 3.3., the analytical body stiffness contribution has remained approximately 90% of total roll stiffness. However, because of the potentially flawed testing method, the contribution of body stiffness cannot be discounted.

### 3.5.5 Prevailing Road Conditions

Yaw overshoot can be affected by road conditions such as extreme road camber, split mu surfaces, temperature differences and general road standards (pot holes, surface deterioration, tram-lines etc). To eliminate these factors, all testing of RBB effects have been performed on a controlled, homogeneous test surface. As the effects can still be determined, road conditions are not considered to be a determining factor in this study.

### 3.5.6 Driver Inputs

As the effect on the vehicle is noticeable by the driver and can be quantified in terms of body response to global stiffness, it can be determined that driver feel and variation is also a contributing factor in this study. However, it should be noted that RBB deletion can be quantified in some instances, Figure 3.12. As these tests were performed with a steering drone (robot), driver variation can be eliminated.

### 3.5.7 Wind

Although prevailing weather conditions can accentuate the issue of yaw response, it has been shown during testing that side wind is not a determining factor in this study.

As all of these factors can be discounted, (except body stiffness), other non-traditional influences, such as vehicle body hysteresis, (Makino, 2004) should also be considered.

## 3.6 Vehicle Body Hysteresis

Studies by Mazda in Japan have suggested that hysteresis rather than global stiffness have more of an effect on vehicle rigidity feel, and that it is this phenomenon which should be used to correlate driver feedback with vehicle performance metrics (Makino, 2004).

An example of this is the 2004 VW Polo which has relatively low global body stiffness compared to its competitors but has a hysteresis value of 4.7% compared to the existing Mazda products of the time that were over 11%, Figure 3.16 (Makino, 2004).

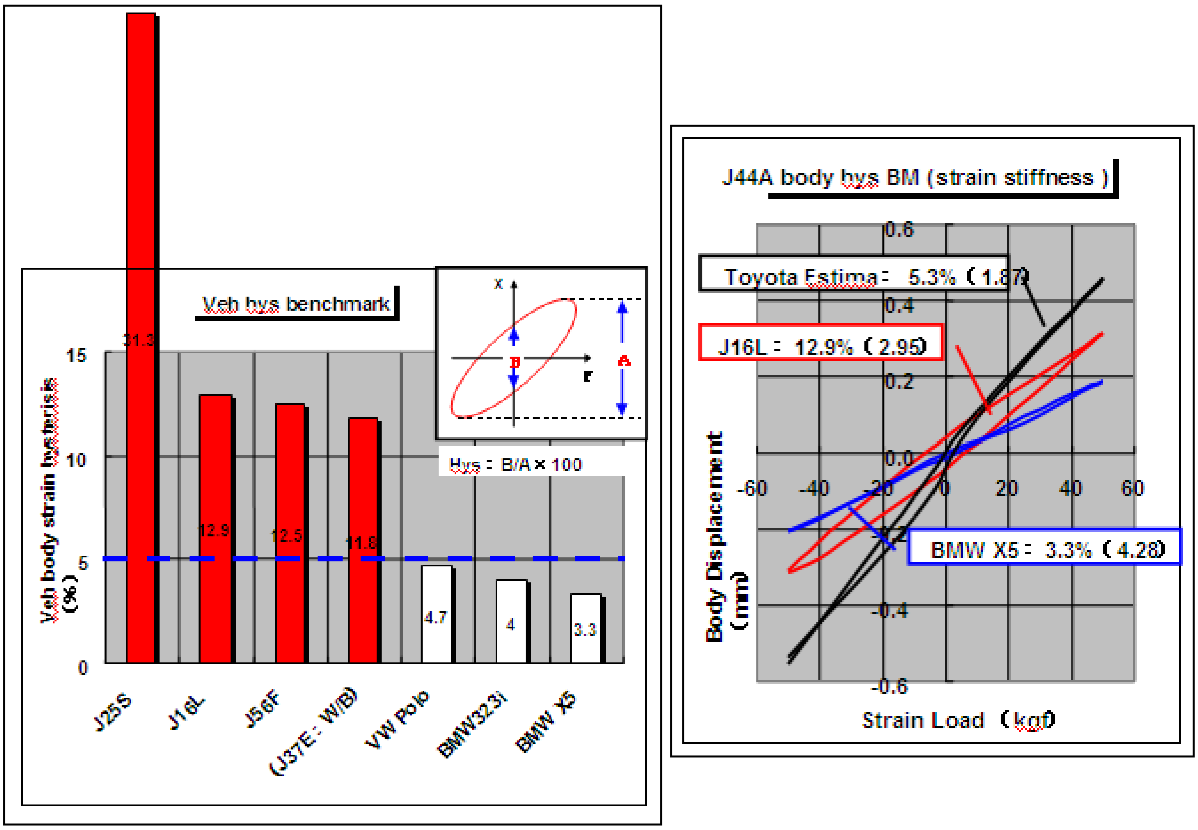



**Figure 3.16** – Hysteresis Comparison (Makino, 2004)

As steel acts as an almost perfect spring in its natural state, it was discovered that stress concentrations at spotwelds and notches in the sheet metal were the main causes of the hysteresis and that these correlated with body stress points that traditionally caused structural issues on durability testing.

Traditionally, areas of high stress are strengthened to ensure durability. However, Mazda state that making a stress point stronger, whilst overcoming durability issues, does not necessarily improve customer perceptions of body rigidity feel.

As can be seen from the results in the study below, correlation is shown to exist between driver responses and hysteresis but not with global stiffness, Figure 3.17 & Appendix A.

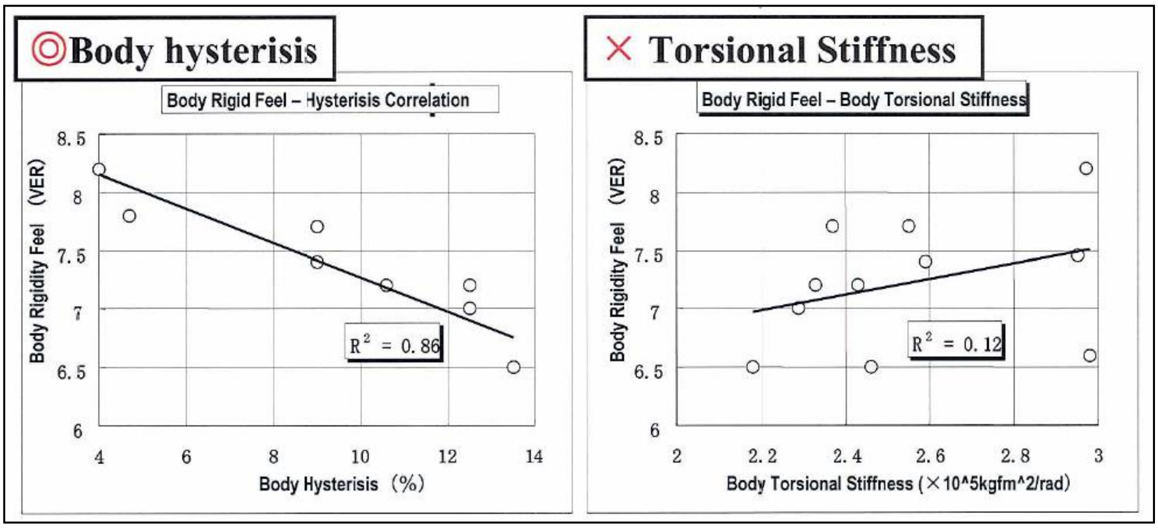



**Figure 3.17** – Body Hysteresis Vs Global Body Stiffness Correlation (Makino, 2004)

Based on this phenomenon, Mazda body hysteresis targets suggest that a 5% maximum target should be set regardless of overall torsional rigidity.

Commercial vehicle hysteresis values are unknown, as tests for this criteria are not widely used outside of Japan. However, stress concentrations of spotwelds and their durability impacts are known and can be evaluated on an FEA model. As these points are also known to impact hysteresis, overall body rigidity feel can be evaluated without the hysteresis value being obtained.

The highest stress concentrations of the body are in the following areas:-

- Strut towers
- Siderails (and outriggers)
- Roof structure (particulary roof load attachment)
- Chassis crossmember attachment points
- RWD propshaft supporting X-members
- Body sides around SLD 'D' ring
- Tailgate hinges, locks and latches
- Hood hinges, and latches
- Grille opening panel
- Upper front door closing panel

The areas around the closures (hood, doors etc) are of particular interest as these clearly indicate high stress points caused by body twisting. Investigations into small passenger cars highlighted similar stress points which indicate common themes around vehicle body design, Figure 3.18 (Makino, 2004).

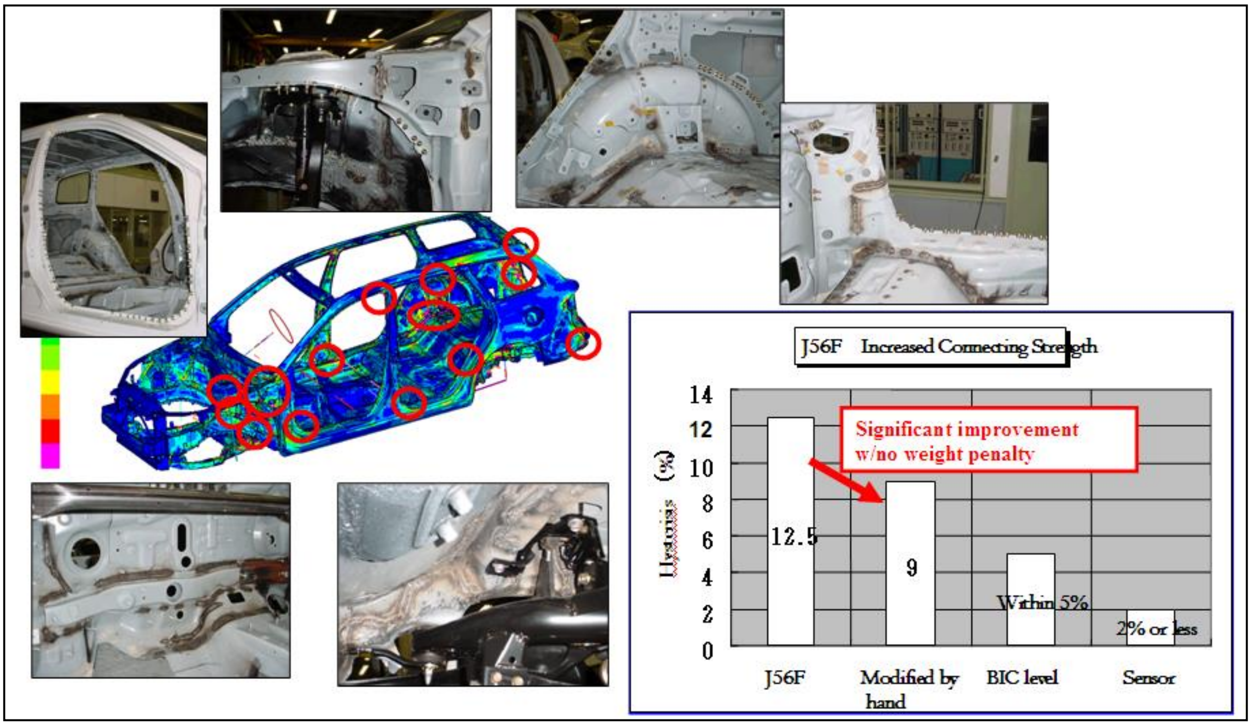



**Figure 3.18** – Typical Vehicle Body High Stress Locations (Makino, 2004)

Simply by removing the high stress concentration points, hysteresis can be reduced significantly without adding weight or affecting global body stiffness. This is desirable for a number of reasons:-

- Vehicle weight determines inertia classes for vehicle taxation
- No extra designed components are required
- Body in white dimensions can be maintained
- Framing fixtures can be maintained
- Interior trim can be packaged without modification
- Process is invisible to the customer

## 3.7 ADAMS Simulation

It should theoretically be possible to simulate the effects of the RBB using an ADAMS model of the full vehicle. However, ADAMS modelling is usually done only on the suspension due to the complexity of the full vehicle model and the time to run any simulations. The body is not simulated in ADAMS as to run effectively, it must be modelled as a flexible element to give the correct responses in transient handling.

The generic ADAMS and CAE modelling consists of:-

- Rigid element modelling for the vehicle body in CAE
- Suspension modelling assuming an infinitely stiff body with simulated loads on each corner in ADAMS

It is therefore clear that to establish any link between the vehicle response with and without the RBB, the ADAMS model must be built using the vehicle body modelled as a flexible element. A full vehicle ADAMS model was commissioned to establish if these links exist; an example of an ADAMS model is shown in Figure 2.9..

To establish the body as a flexible element, the inertia properties of the body finite element model must be adjusted. Body mass and centre of gravity positions were primarily adjusted by adding non-structural masses such as:-

- Engine
- Seats
- Fuel tank
- Driveline
- Doors etc

Moments of inertia were also adjusted slightly until the mass and CoG of the flexible body met the values of the ADAMS rigid body.

This had the effect of lowering the moments of inertia, especially in the *Ixx* direction, Table 3.7.

| | | | | rigid Body | | | flex Body w/o RBB | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Mass (kg) | | | | | 1524,5 | | | 1524,4 | | +0,1 kg | | |
| COG coordinates (mm) | X | | | | 3244,4 | | | 3247,0 | | -2,7 mm | | |
| | Y | | | | 1,7 | | | -2,8 | | +4,5 mm | | |
| | Z | | | | 1784,4 | | | 1786,8 | | -2,5 mm | | |
| Inertia tensor (to.mm^4) | Ixx | Ixy | Ixz | 9,4E+05 | 0 | 0 | 7,5E+05 | -1,4E+03 | -2,5E+05 | -21,0% | - | - |
| | Ixy | Iyy | Iyz | 0 | 4,2E+06 | 0 | -1,4E+03 | 3,9E+06 | -4,3E+03 | - | -7,8% | - |
| | Ixz | Iyz | Izz | 0 | 0 | 4,2E+06 | -2,5E+05 | -4,3E+03 | 4,0E+06 | - | - | -3,6% |

**Table 3.7** – Rigid Body vs Flexible Body Analysis in ADAMS

To test the models in ADAMS for integrity, a double lane change manoeuvre was performed in ADAMS comparing the responses of the rigid and flexible body model.

The double lane change is a standard procedure at a calculated speed of 100kph on the course shown in Figure 3.19.

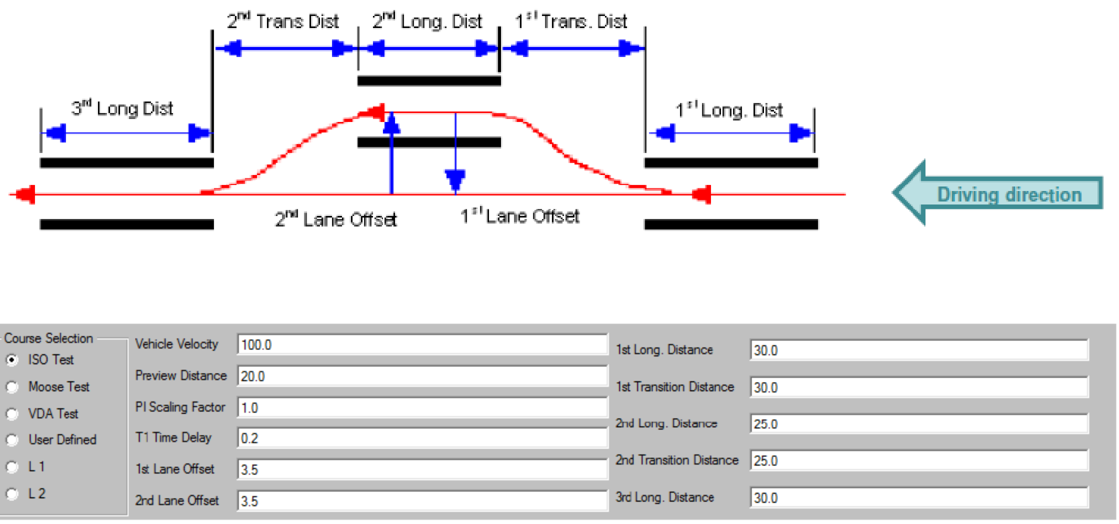



**Figure 3.19** – Double Lane Change Procedure in ADAMS

Steering wheel moments and vehicle path analysis are shown in Figure 3.20, that demonstrate the good correlation of the flexible body and the rigid body.

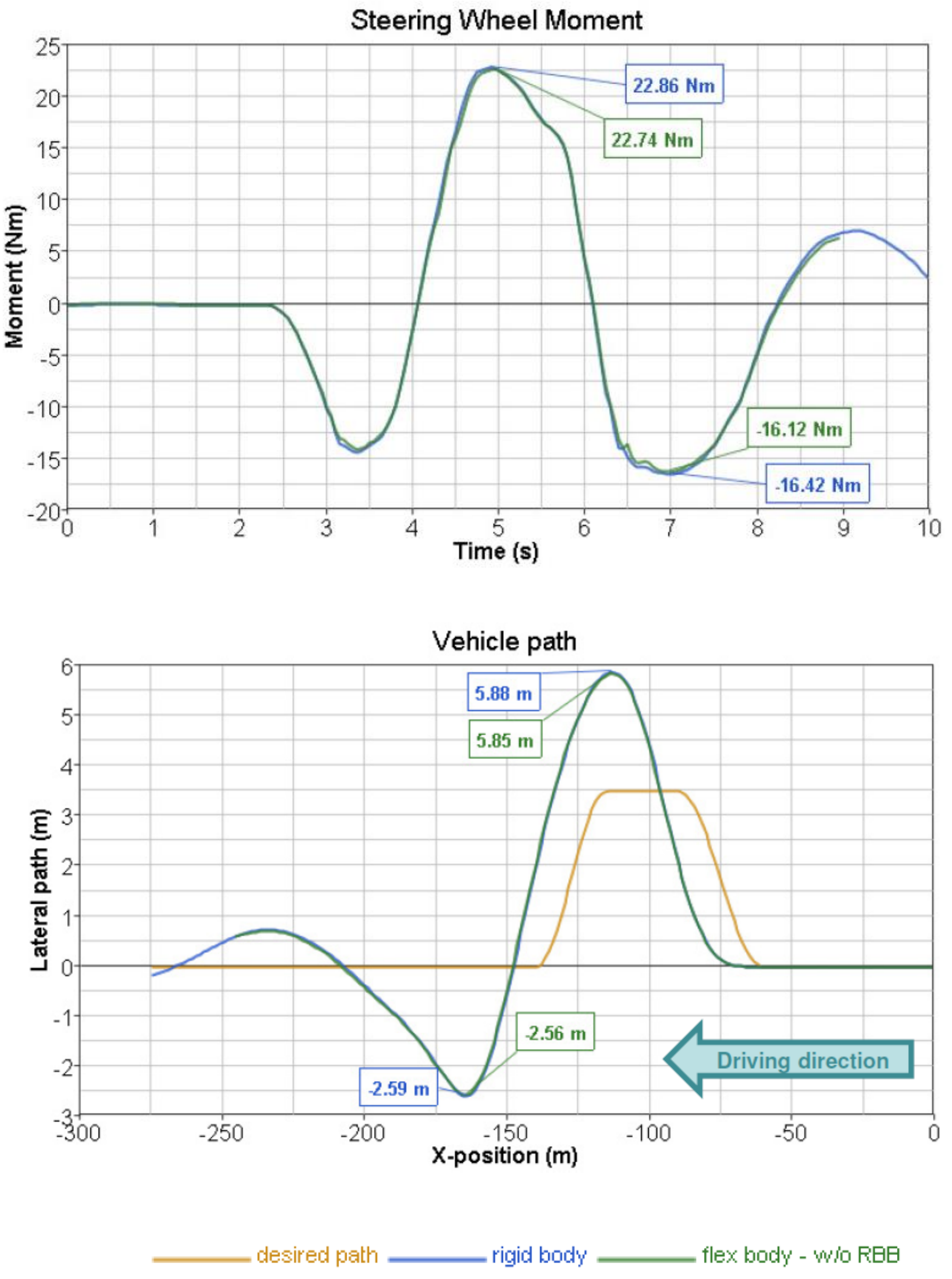



**Figure 3.20** – ADAMS Correlation Results for Rigid Vs Flexible Body Simulation

### 3.7.1 Results From ADAMS Analysis with Flexible Body Model

The following metrics were analysed during the analysis of the flexible vs rigid model:-

- Vehicle Path: *Y* Position vs *X* Position
- Steering Wheel Moment
- Toe, Camber & Caster
- Lateral Acceleration
- Yaw Rate
- Roll Angle
- Side Slip Angle

Yaw rate describes the angular velocity around the *Z* axis of the vehicle co-ordinate system, Figure 3.21.

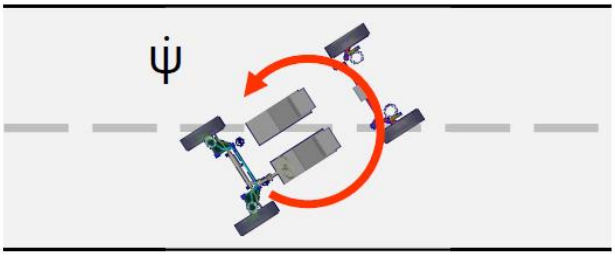


**Figure 3.21** – Yaw Rate

Roll angle describes the angle around the *X* axis between the vehicle Z direction and the global Z direction, Figure 3.22.

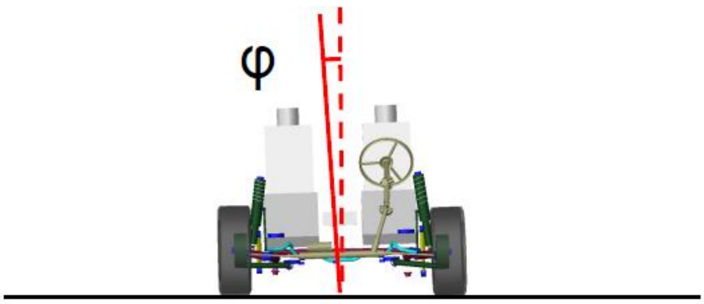


**Figure 3.22** – Roll Angle

Side slip angle describes the angle between the vehicle *X* direction and the direction of the vehicle velocity vector. Tyre side slip angle describes the angle between the rolling wheel's velocity vector and the direction towards which it is pointing, Figure 3.23.

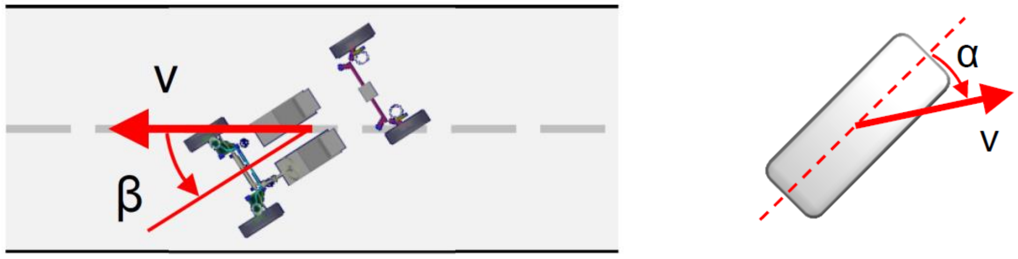



**Figure 3.23** – Side Slip Angle and Tyre Slip Angle

### 3.7.2 Influence of Rear Bumper Beam on Flexible Body ADAMS Model

As the RBB is added to the body on the rear end of the vehicle, the COG is located more towards the rear of the vehicle. This leads to slightly higher *YY* and *ZZ* moments of inertia, even though a static finite element analysis showed that the bodies have the same roll stiffness, Figure 3.24.

| | | | | flex Body w/o RBB | | | flex Body with RBB | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Mass (kg) | | | | 1524,4 | | | 1531,2 | | | +6,8 kg | | |
| COG coordinates (mm) | X | | | 3247,0 | | | 3260,2 | | | +13,2 mm | | |
| | Y | | | -2,8 | | | -2,8 | | | +0,0 mm | | |
| | Z | | | 1786,8 | | | 1784,6 | | | -2,2 mm | | |
| Inertia tensor (to.mm^4) | Ixx | Ixy | Ixz | 7,5E+05 | -1,4E+03 | -2,5E+05 | 7,5E+05 | -1,5E+03 | -2,4E+05 | +0,4% | +3,9% | -4,1% |
| | Ixy | Iyy | Iyz | -1,4E+03 | 3,9E+06 | -4,3E+03 | -1,5E+03 | 4,0E+06 | -4,3E+03 | +3,9% | +1,6% | -0,3% |
| | Ixz | Iyz | Izz | -2,5E+05 | -4,3E+03 | 4,0E+06 | -2,4E+05 | -4,3E+03 | 4,1E+06 | -4,1% | -0,3% | +1,5% |

**Figure 3.24** – Flexible Body With and Without RBB

Flexible bodies are described by their normal modes. Mode frequencies of both bodies can therefore be compared. Of the first 146 modes analysed, it was noted that 26 deviate by over 5 Hz which is a significant difference, Table 3.8.

| mode | mode frequency (Hz) | | difference (Hz) | mode | mode frequency (Hz) | | difference (Hz) | mode | mode frequency (Hz) | | difference (Hz) | mode | mode frequency (Hz) | | difference (Hz) |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | with RBB | without RBB | | | with RBB | without RBB | | | with RBB | without RBB | | | with RBB | without RBB | |
| 7 | 8,44 | 8,44 | +0,0 | 42 | 72,76 | 72,79 | +0,0 | 77 | 326,02 | 332,78 | +6,8 | 112 | 1140,47 | 1180,30 | +39,8 |
| 8 | 8,73 | 8,73 | +0,0 | 43 | 75,51 | 75,59 | +0,1 | 78 | 346,53 | 348,88 | +2,3 | 113 | 1193,49 | 1197,56 | +4,1 |
| 9 | 14,95 | 14,95 | +0,0 | 44 | 81,48 | 81,62 | +0,1 | 79 | 363,16 | 367,98 | +4,8 | 114 | 1316,36 | 1317,33 | +1,0 |
| 10 | 15,52 | 15,52 | +0,0 | 45 | 86,60 | 87,24 | +0,6 | 80 | 384,20 | 384,72 | +0,5 | 115 | 1421,80 | 1422,00 | +0,2 |
| 11 | 17,43 | 17,43 | +0,0 | 46 | 90,75 | 90,77 | +0,0 | 81 | 400,30 | 397,52 | -2,8 | 116 | 1463,57 | 1466,56 | +3,0 |
| 12 | 17,46 | 17,46 | +0,0 | 47 | 93,04 | 93,17 | +0,1 | 82 | 405,03 | 434,76 | +29,7 | 117 | 1594,72 | 1595,01 | +0,3 |
| 13 | 17,52 | 17,52 | +0,0 | 48 | 96,93 | 97,07 | +0,1 | 83 | 430,63 | 443,39 | +12,8 | 118 | 1659,93 | 1661,65 | +1,7 |
| 14 | 18,43 | 18,47 | +0,0 | 49 | 99,75 | 99,78 | +0,0 | 84 | 437,00 | 445,58 | +8,6 | 119 | 1738,69 | 1745,31 | +6,6 |
| 15 | 18,94 | 18,95 | +0,0 | 50 | 112,55 | 112,63 | +0,1 | 85 | 446,06 | 454,73 | +8,7 | 120 | 1751,38 | 1755,37 | +4,0 |
| 16 | 19,42 | 19,43 | +0,0 | 51 | 117,78 | 117,85 | +0,1 | 86 | 458,79 | 462,75 | +4,0 | 121 | 2034,35 | 2023,24 | -11,1 |
| 17 | 20,43 | 20,49 | +0,1 | 52 | 125,27 | 125,36 | +0,1 | 87 | 469,12 | 480,91 | +11,8 | 122 | 2090,24 | 2167,98 | +77,7 |
| 18 | 22,03 | 22,06 | +0,0 | 53 | 127,13 | 127,55 | +0,4 | 88 | 484,37 | 487,78 | +3,4 | 123 | 3026,66 | 3028,45 | +1,8 |
| 19 | 22,62 | 22,63 | +0,0 | 54 | 128,42 | 128,61 | +0,2 | 89 | 499,57 | 499,52 | -0,0 | 124 | 3069,71 | 3070,39 | +0,7 |
| 20 | 24,83 | 24,97 | +0,1 | 55 | 135,49 | 135,88 | +0,4 | 90 | 525,80 | 526,13 | +0,3 | 125 | 3166,16 | 3166,52 | +0,4 |
| 21 | 25,64 | 25,65 | +0,0 | 56 | 136,70 | 137,11 | +0,4 | 91 | 533,79 | 541,61 | +7,8 | 126 | 3173,43 | 3173,36 | -0,1 |
| 22 | 26,50 | 26,51 | +0,0 | 57 | 145,40 | 147,12 | +1,7 | 92 | 547,71 | 552,23 | +4,5 | 127 | 3436,04 | 3437,22 | +1,2 |
| 23 | 27,66 | 27,66 | -0,0 | 58 | 146,26 | 147,49 | +1,2 | 93 | 556,22 | 560,83 | +4,6 | 128 | 3536,39 | 3538,91 | +2,5 |
| 24 | 28,62 | 28,63 | +0,0 | 59 | 153,77 | 153,85 | +0,1 | 94 | 575,23 | 576,12 | +0,9 | 129 | 3909,63 | 3909,92 | +0,3 |
| 25 | 29,55 | 29,59 | +0,0 | 60 | 163,35 | 163,54 | +0,2 | 95 | 603,48 | 602,18 | -1,3 | 130 | 4049,41 | 4049,93 | +0,5 |
| 26 | 30,25 | 30,26 | +0,0 | 61 | 171,15 | 171,16 | +0,0 | 96 | 621,44 | 623,48 | +2,0 | 131 | 5956,32 | 5958,40 | +2,1 |
| 27 | 30,44 | 30,44 | +0,0 | 62 | 178,83 | 178,83 | +0,0 | 97 | 697,53 | 697,83 | +0,3 | 132 | 6465,79 | 6465,87 | +0,1 |
| 28 | 30,67 | 30,67 | -0,0 | 63 | 184,14 | 187,89 | +3,8 | 98 | 727,42 | 728,53 | +1,1 | 133 | 6472,41 | 6472,41 | +0,0 |
| 29 | 31,85 | 31,84 | -0,0 | 64 | 204,13 | 205,11 | +1,0 | 99 | 763,02 | 788,64 | +25,6 | 134 | 6843,69 | 6850,55 | +6,9 |
| 30 | 34,11 | 34,11 | +0,0 | 65 | 223,00 | 223,66 | +0,7 | 100 | 788,15 | 798,47 | +10,3 | 135 | 7931,19 | 7931,55 | +0,4 |
| 31 | 38,38 | 38,56 | +0,2 | 66 | 235,01 | 235,32 | +0,3 | 101 | 803,79 | 811,24 | +7,4 | 136 | 7982,54 | 7983,91 | +1,4 |
| 32 | 40,71 | 40,77 | +0,1 | 67 | 248,70 | 251,19 | +2,5 | 102 | 811,84 | 817,90 | +6,1 | 137 | 12466,75 | 12466,76 | +0,0 |
| 33 | 46,64 | 46,91 | +0,3 | 68 | 253,44 | 254,56 | +1,1 | 103 | 828,17 | 831,62 | +3,5 | 138 | 12493,07 | 12493,07 | +0,0 |
| 34 | 47,29 | 47,30 | +0,0 | 69 | 261,58 | 268,35 | +6,8 | 104 | 836,05 | 836,78 | +0,7 | 139 | 22275,23 | 22275,24 | +0,0 |
| 35 | 52,75 | 52,85 | +0,1 | 70 | 267,45 | 275,58 | +8,1 | 105 | 873,81 | 889,48 | +15,7 | 140 | 22322,68 | 22322,69 | +0,0 |
| 36 | 55,30 | 55,32 | +0,0 | 71 | 276,32 | 277,37 | +1,0 | 106 | 898,35 | 904,50 | +6,2 | 141 | 37906,16 | 37906,27 | +0,1 |
| 37 | 59,46 | 58,72 | -0,7 | 72 | 297,09 | 300,38 | +3,3 | 107 | 920,43 | 932,09 | +11,7 | 142 | 37950,75 | 37950,76 | +0,0 |
| 38 | 61,83 | 61,86 | +0,0 | 73 | 300,47 | 304,24 | +3,8 | 108 | 991,75 | 989,37 | -2,4 | 143 | 42641,55 | 42641,60 | +0,1 |
| 39 | 65,73 | 65,59 | -0,1 | 74 | 301,74 | 320,45 | +18,7 | 109 | 1027,49 | 1072,55 | +45,1 | 144 | 42675,93 | 42675,95 | +0,0 |
| 40 | 65,97 | 66,14 | +0,2 | 75 | 313,03 | 325,69 | +12,7 | 110 | 1063,47 | 1078,82 | +15,3 | 145 | 49364,19 | 49364,24 | +0,1 |
| 41 | 71,84 | 72,03 | +0,2 | 76 | 323,58 | 328,12 | +4,5 | 111 | 1077,38 | 1104,82 | +27,4 | 146 | 49400,39 | 49400,40 | +0,0 |

**Table 3.8** – Body Mode Frequency Comparison

Comparing the steering wheel moments and vehicle path graphs with and without RBB, it can be seen that significant differences are evident in the vehicle body responses, Figure 3.25.

With the RBB, the driver must apply a higher moment at the steering wheel; 0.2 and 0.3 Nm of torque.

The analysis also shows that the model with the RBB deviates by 150mm in the Y direction in the second lane change than the model without the RBB.

These results are consistent on all of the measurement results are shown in Appendix B.

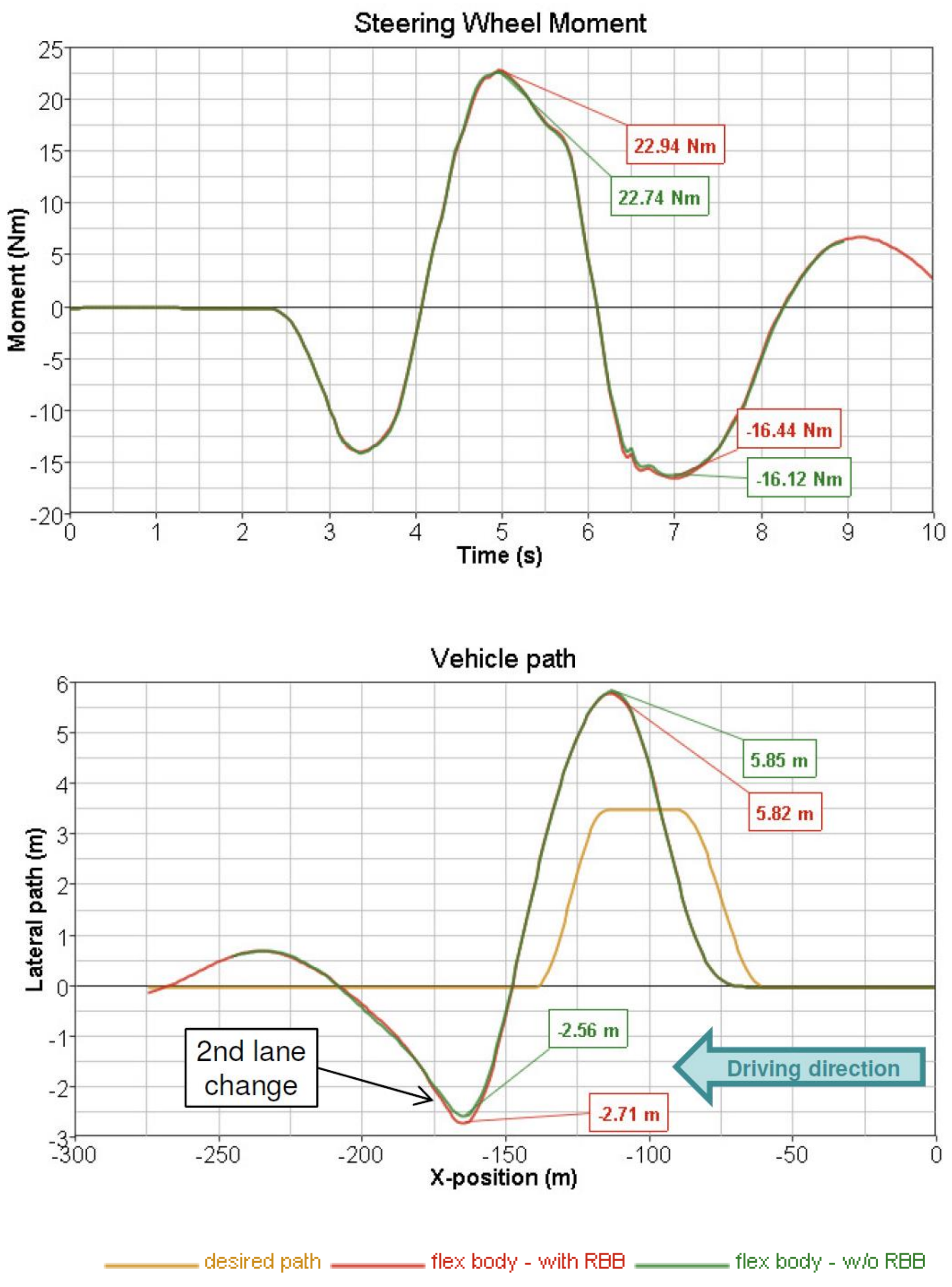



**Figure 3.25** – RBB Influence on Vehicle Response During Double Lane Change Event

The differences show that the vehicle responses are slightly better **without** the RBB fitted – which is contrary to the initial report of the vehicles from the driver perspective.

The explanation for this could be that even though drivers complain of deteriorated steering feel, this is because the steering is now too sharp which means that drivers are over compensating with steering inputs making the perception different to the reality.

The next step to investigate this phenomenon in the ADAMS model is to run the simulation using the same frequency range as a driver; 0 – 3 Hz. This would span a more dynamic range of events rather than just a double lane change manoeuvre.

## 3.8 Summary

As seen from the testing reviews, a number of factors could be seen as significant.

The vehicle response effects could be objectively measured using:-

1. Simple vehicle bend test by the jacking of a single wheel
2. Understeer gradient during transit swept steer on a vehicle test
3. Vehicle steer path during double lane change manoeuvres using a full vehicle ADAMS model

Body hysteresis, whilst not considered in physical testing at this stage, could also be significant as the RBB function may be able to be replicated in the body stiffness.

The testing results discussion will focus on changing the body stiffness in the ADAMS model to test for body stiffness improvements that could nullify the effects of the RBB.

# 4. Discussion of Results

## 4.1 Introduction

The following chapter will discuss the results of the testing and draw conclusions to explain the analytical results and the physical measured data. Further suggestions regarding using the ADAMS model to improve the vehicle to reduce the effects of the RBB presence will be discussed.

Body hysteresis and stiffness improvements will be used for the results conclusion.

A discussion will also be written around the incompatibility of the driver reports on the vehicle with and without RBB versus the ADAMS model prediction.

## 4.2 Analytical Body Stiffness and Hysteresis Improvements

As could be seen from the study into body stiffness hysteresis in 3.6, it is possible that the RBB deletion is influencing this phenomenon. To establish this on a physical test would entail many weeks of preparation and data analysis as well as re-testing any alternative solutions.

Due to the time and resources required to establish the physical body hysteresis, the full vehicle CAE model can be analysed to determine the key areas to replicate the modes that simulate the presence of the RBB. This should be analysed with a view to addressing known high stress points in the vehicle body to improve the overall hysteresis, as studies show that this has the effect of improving rigidity perception (Makino, 2004).

Traditional weak spots in the body sheet metal established during testing are:-

- A-pillars
- B-pillar and door closure interfaces (up to the roof line)
- C-pillars
- Roof rails
- Underbody attachment at rear jounce bumpers
- Rear anti-roll bar attachment

Improvements in the CAE model could be fed back into the ADAMS simulation to determine the effects of the body stiffness to the transient handling response.

This would give a positive indication that the body hysteresis is a controlling factor of steering response. This would also serve as durability improvements to the vehicle at potentially minimal cost.

Body improvements show that the greatest influence in the body stiffness are around the A pillars, B pillars, C pillars and the roof rails.  These have by far the most impact to the way the parametric ADAMS model behaves on the lane change manoeuvre.  Not all vehicles are fitted with rear anti-roll bars and so this can be ruled out as a significant influence.

Likewise, the body stiffness at the rear jounce bumpers, although reacting a great deal of load (upwards of 50kN on some vehicles), do not have any strength issues on lighter vehicles even though the RBB phenomenon is still prevalent.

The body was strengthened analytically as described by assuming E-modulus values multiplied by 1000, Figure 4.1.

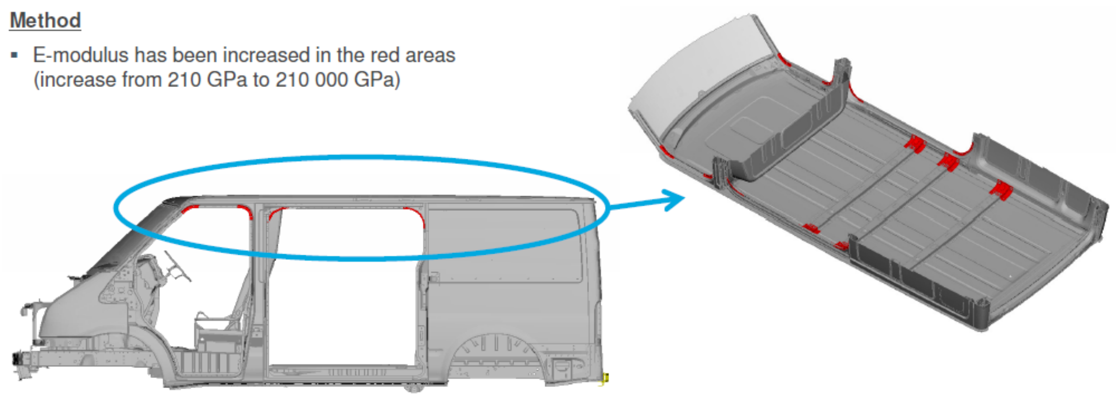



**Figure 4.1** – Body Stiffness Increase Locations

Describing the body by its normal modes, the first 146 modes can again be analysed.  This study showed an increase in modes that deviate by over 5 Hz, compared to 26 modes prior to body strengthening actions. These are shown with and without RBB comparing stiffened vs non-stiffened bodies, Tables 4.1 & 4.2.

| mode | mode frequency (Hz) | | difference | mode | mode frequency (Hz) | | difference | mode | mode frequency (Hz) | | difference | mode | mode frequency (Hz) | | difference |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | w/o RBB | stiffened w/o RBB | (Hz) | | w/o RBB | stiffened w/o RBB | (Hz) | | w/o RBB | stiffened w/o RBB | (Hz) | | w/o RBB | stiffened w/o RBB | (Hz) |
| 7 | 8,44 | 8,44 | +0,0 | 42 | 72,79 | 72,90 | +0,1 | 77 | 326,02 | 340,00 | +14,0 | 112 | 1140,47 | 1190,00 | +49,5 |
| 8 | 8,73 | 8,73 | -0,0 | 43 | 75,59 | 75,60 | +0,0 | 78 | 346,53 | 350,00 | +3,5 | 113 | 1193,49 | 1220,00 | +26,5 |
| 9 | 14,95 | 14,90 | -0,0 | 44 | 81,62 | 81,40 | -0,2 | 79 | 363,16 | 376,00 | +12,8 | 114 | 1316,36 | 1340,00 | +23,6 |
| 10 | 15,52 | 15,50 | -0,0 | 45 | 87,24 | 86,20 | -1,0 | 80 | 384,20 | 387,00 | +2,8 | 115 | 1421,80 | 1430,00 | +8,2 |
| 11 | 17,43 | 17,40 | -0,0 | 46 | 90,77 | 90,50 | -0,3 | 81 | 400,30 | 400,00 | -0,3 | 116 | 1463,57 | 1470,00 | +6,4 |
| 12 | 17,46 | 17,50 | +0,0 | 47 | 93,17 | 93,20 | +0,0 | 82 | 405,03 | 439,00 | +34,0 | 117 | 1594,72 | 1620,00 | +25,3 |
| 13 | 17,52 | 17,50 | -0,0 | 48 | 97,07 | 98,30 | +1,2 | 83 | 430,63 | 445,00 | +14,4 | 118 | 1659,93 | 1670,00 | +10,1 |
| 14 | 18,47 | 19,10 | +0,6 | 49 | 99,78 | 100,00 | +0,2 | 84 | 437,00 | 455,00 | +18,0 | 119 | 1738,69 | 1750,00 | +11,3 |
| 15 | 18,95 | 19,20 | +0,3 | 50 | 112,63 | 112,00 | -0,6 | 85 | 446,06 | 459,00 | +12,9 | 120 | 1751,38 | 1760,00 | +8,6 |
| 16 | 19,43 | 20,10 | +0,7 | 51 | 117,85 | 118,00 | +0,1 | 86 | 458,79 | 470,00 | +11,2 | 121 | 2034,35 | 2030,00 | -4,4 |
| 17 | 20,49 | 21,40 | +0,9 | 52 | 125,36 | 126,00 | +0,6 | 87 | 469,12 | 483,00 | +13,9 | 122 | 2090,24 | 2170,00 | +79,8 |
| 18 | 22,06 | 22,50 | +0,4 | 53 | 127,55 | 128,00 | +0,5 | 88 | 484,37 | 498,00 | +13,6 | 123 | 3026,66 | 3030,00 | +3,3 |
| 19 | 22,63 | 23,00 | +0,4 | 54 | 128,61 | 129,00 | +0,4 | 89 | 499,57 | 509,00 | +9,4 | 124 | 3069,71 | 3070,00 | +0,3 |
| 20 | 24,97 | 25,40 | +0,4 | 55 | 135,88 | 135,00 | -0,9 | 90 | 525,80 | 540,00 | +14,2 | 125 | 3166,16 | 3170,00 | +3,8 |
| 21 | 25,65 | 26,30 | +0,7 | 56 | 137,11 | 138,00 | +0,9 | 91 | 533,79 | 547,00 | +13,2 | 126 | 3173,43 | 3240,00 | +66,6 |
| 22 | 26,51 | 26,60 | +0,1 | 57 | 147,12 | 148,00 | +0,9 | 92 | 547,71 | 555,00 | +7,3 | 127 | 3436,04 | 3440,00 | +4,0 |
| 23 | 27,66 | 27,90 | +0,2 | 58 | 147,49 | 148,00 | +0,5 | 93 | 556,22 | 575,00 | +18,8 | 128 | 3536,39 | 3540,00 | +3,6 |
| 24 | 28,63 | 28,80 | +0,2 | 59 | 153,85 | 154,00 | +0,1 | 94 | 575,23 | 586,00 | +10,8 | 129 | 3909,63 | 3970,00 | +60,4 |
| 25 | 29,59 | 29,10 | -0,5 | 60 | 163,54 | 164,00 | +0,5 | 95 | 603,48 | 607,00 | +3,5 | 130 | 4049,41 | 4240,00 | +190,6 |
| 26 | 30,26 | 30,40 | +0,1 | 61 | 171,16 | 173,00 | +1,8 | 96 | 621,44 | 660,00 | +38,6 | 131 | 5956,32 | 5960,00 | +3,7 |
| 27 | 30,44 | 30,50 | +0,1 | 62 | 178,83 | 181,00 | +2,2 | 97 | 697,53 | 701,00 | +3,5 | 132 | 6465,79 | 6470,00 | +4,2 |
| 28 | 30,67 | 30,70 | +0,0 | 63 | 187,89 | 186,00 | -1,9 | 98 | 727,42 | 730,00 | +2,6 | 133 | 6472,41 | 6470,00 | -2,4 |
| 29 | 31,84 | 32,10 | +0,3 | 64 | 205,11 | 207,00 | +1,9 | 99 | 763,02 | 789,00 | +26,0 | 134 | 6843,69 | 6850,00 | +6,3 |
| 30 | 34,11 | 34,10 | -0,0 | 65 | 223,66 | 227,00 | +3,3 | 100 | 788,15 | 799,00 | +10,9 | 135 | 7931,19 | 7930,00 | -1,2 |
| 31 | 38,56 | 40,30 | +1,7 | 66 | 235,32 | 235,00 | -0,3 | 101 | 803,79 | 812,00 | +8,2 | 136 | 7982,54 | 7980,00 | -2,5 |
| 32 | 40,77 | 45,00 | +4,2 | 67 | 251,19 | 252,00 | +0,8 | 102 | 811,84 | 821,00 | +9,2 | 137 | 12466,75 | 12500,00 | +33,2 |
| 33 | 46,91 | 45,70 | -1,2 | 68 | 254,56 | 259,00 | +4,4 | 103 | 828,17 | 834,00 | +5,8 | 138 | 12493,07 | 12500,00 | +6,9 |
| 34 | 47,30 | 48,90 | +1,6 | 69 | 268,35 | 268,00 | -0,4 | 104 | 836,05 | 844,00 | +7,9 | 139 | 22275,23 | 22300,00 | +24,8 |
| 35 | 52,85 | 51,60 | -1,3 | 70 | 275,58 | 277,00 | +1,4 | 105 | 873,81 | 893,00 | +19,2 | 140 | 22322,68 | 22300,00 | -22,7 |
| 36 | 55,32 | 57,20 | +1,9 | 71 | 277,37 | 278,00 | +0,6 | 106 | 898,35 | 932,00 | +33,6 | 141 | 37906,16 | 37900,00 | -6,2 |
| 37 | 58,72 | 58,70 | -0,0 | 72 | 300,38 | 300,00 | -0,4 | 107 | 920,43 | 958,00 | +37,6 | 142 | 37950,75 | 38000,00 | +49,2 |
| 38 | 61,86 | 63,10 | +1,2 | 73 | 304,24 | 307,00 | +2,8 | 108 | 991,75 | 992,00 | +0,2 | 143 | 42641,55 | 42600,00 | -41,5 |
| 39 | 65,59 | 66,00 | +0,4 | 74 | 320,45 | 323,00 | +2,5 | 109 | 1027,49 | 1080,00 | +52,5 | 144 | 42675,93 | 42700,00 | +24,1 |
| 40 | 66,14 | 67,80 | +1,7 | 75 | 325,69 | 328,00 | +2,3 | 110 | 1063,47 | 1100,00 | +36,5 | 145 | 49364,19 | 49400,00 | +35,8 |
| 41 | 72,03 | 72,50 | +0,5 | 76 | 328,12 | 332,00 | +3,9 | 111 | 1077,38 | 1120,00 | +42,6 | 146 | 49400,39 | 49400,00 | -0,4 |

**Table 4.1** – Body Mode Frequency Comparison – Without RBB

| mode | mode frequency (Hz) | | difference | mode | mode frequency (Hz) | | difference | mode | mode frequency (Hz) | | difference | mode | mode frequency (Hz) | | difference |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | with RBB | stiffened with RBB | (Hz) | | with RBB | stiffened with RBB | (Hz) | | with RBB | stiffened with RBB | (Hz) | | with RBB | stiffened with RBB | (Hz) |
| 7 | 8,44 | 8,44 | +0,0 | 42 | 72,76 | 72,85 | +0,1 | 77 | 326,02 | 340,39 | +14,4 | 112 | 1140,47 | 1194,03 | +53,6 |
| 8 | 8,73 | 8,73 | +0,0 | 43 | 75,51 | 75,58 | +0,1 | 78 | 346,53 | 350,21 | +3,7 | 113 | 1193,49 | 1221,27 | +27,8 |
| 9 | 14,95 | 14,95 | +0,0 | 44 | 81,48 | 81,35 | -0,1 | 79 | 363,16 | 376,08 | +12,9 | 114 | 1316,36 | 1339,78 | +23,4 |
| 10 | 15,52 | 15,53 | +0,0 | 45 | 86,60 | 86,18 | -0,4 | 80 | 384,20 | 387,01 | +2,8 | 115 | 1421,80 | 1426,37 | +4,6 |
| 11 | 17,43 | 17,44 | +0,0 | 46 | 90,75 | 90,47 | -0,3 | 81 | 400,30 | 399,82 | -0,5 | 116 | 1463,57 | 1466,92 | +3,4 |
| 12 | 17,46 | 17,47 | +0,0 | 47 | 93,04 | 93,18 | +0,1 | 82 | 405,03 | 439,16 | +34,1 | 117 | 1594,72 | 1623,35 | +28,6 |
| 13 | 17,52 | 17,53 | +0,0 | 48 | 96,93 | 98,30 | +1,4 | 83 | 430,63 | 444,90 | +14,3 | 118 | 1659,93 | 1669,24 | +9,3 |
| 14 | 18,43 | 19,06 | +0,6 | 49 | 99,75 | 100,15 | +0,4 | 84 | 437,00 | 454,68 | +17,7 | 119 | 1738,69 | 1747,46 | +8,8 |
| 15 | 18,94 | 19,23 | +0,3 | 50 | 112,55 | 112,39 | -0,2 | 85 | 446,06 | 458,73 | +12,7 | 120 | 1751,38 | 1761,46 | +10,1 |
| 16 | 19,42 | 20,06 | +0,6 | 51 | 117,78 | 118,38 | +0,6 | 86 | 458,79 | 469,99 | +11,2 | 121 | 2034,35 | 2026,89 | -7,5 |
| 17 | 20,43 | 21,45 | +1,0 | 52 | 125,27 | 125,56 | +0,3 | 87 | 469,12 | 483,23 | +14,1 | 122 | 2090,24 | 2168,17 | +77,9 |
| 18 | 22,03 | 22,46 | +0,4 | 53 | 127,13 | 128,16 | +1,0 | 88 | 484,37 | 497,96 | +13,6 | 123 | 3026,66 | 3033,80 | +7,1 |
| 19 | 22,62 | 23,03 | +0,4 | 54 | 128,42 | 129,03 | +0,6 | 89 | 499,57 | 509,27 | +9,7 | 124 | 3069,71 | 3074,86 | +5,2 |
| 20 | 24,83 | 25,39 | +0,6 | 55 | 135,49 | 135,47 | -0,0 | 90 | 525,80 | 540,35 | +14,5 | 125 | 3166,16 | 3169,44 | +3,3 |
| 21 | 25,64 | 26,29 | +0,6 | 56 | 136,70 | 137,62 | +0,9 | 91 | 533,79 | 547,29 | +13,5 | 126 | 3173,43 | 3242,72 | +69,3 |
| 22 | 26,50 | 26,58 | +0,1 | 57 | 145,40 | 147,58 | +2,2 | 92 | 547,71 | 554,94 | +7,2 | 127 | 3436,04 | 3437,34 | +1,3 |
| 23 | 27,66 | 27,92 | +0,3 | 58 | 146,26 | 147,68 | +1,4 | 93 | 556,22 | 575,46 | +19,2 | 128 | 3536,39 | 3539,03 | +2,6 |
| 24 | 28,62 | 28,78 | +0,2 | 59 | 153,77 | 154,21 | +0,4 | 94 | 575,23 | 586,48 | +11,3 | 129 | 3909,63 | 3971,49 | +61,9 |
| 25 | 29,55 | 29,12 | -0,4 | 60 | 163,35 | 164,20 | +0,8 | 95 | 603,48 | 607,32 | +3,8 | 130 | 4049,41 | 4235,92 | +186,5 |
| 26 | 30,25 | 30,41 | +0,2 | 61 | 171,15 | 172,73 | +1,6 | 96 | 621,44 | 659,73 | +38,3 | 131 | 5956,32 | 5958,38 | +2,1 |
| 27 | 30,44 | 30,53 | +0,1 | 62 | 178,83 | 180,94 | +2,1 | 97 | 697,53 | 701,15 | +3,6 | 132 | 6465,79 | 6466,21 | +0,4 |
| 28 | 30,67 | 30,69 | +0,0 | 63 | 184,14 | 185,90 | +1,8 | 98 | 727,42 | 730,13 | +2,7 | 133 | 6472,41 | 6472,26 | -0,1 |
| 29 | 31,85 | 32,13 | +0,3 | 64 | 204,13 | 206,68 | +2,5 | 99 | 763,02 | 789,16 | +26,1 | 134 | 6843,69 | 6850,77 | +7,1 |
| 30 | 34,11 | 34,13 | +0,0 | 65 | 223,00 | 226,76 | +3,8 | 100 | 788,15 | 799,01 | +10,9 | 135 | 7931,19 | 7931,65 | +0,5 |
| 31 | 38,38 | 40,34 | +2,0 | 66 | 235,01 | 235,32 | +0,3 | 101 | 803,79 | 811,93 | +8,1 | 136 | 7982,54 | 7984,04 | +1,5 |
| 32 | 40,71 | 44,98 | +4,3 | 67 | 248,70 | 252,25 | +3,5 | 102 | 811,84 | 820,92 | +9,1 | 137 | 12466,75 | 12467,41 | +0,7 |
| 33 | 46,64 | 45,71 | -0,9 | 68 | 253,44 | 258,87 | +5,4 | 103 | 828,17 | 834,24 | +6,1 | 138 | 12493,07 | 12494,22 | +1,2 |
| 34 | 47,29 | 48,87 | +1,6 | 69 | 261,58 | 268,17 | +6,6 | 104 | 836,05 | 844,33 | +8,3 | 139 | 22275,23 | 22277,60 | +2,4 |
| 35 | 52,75 | 51,59 | -1,2 | 70 | 267,45 | 277,24 | +9,8 | 105 | 873,81 | 893,48 | +19,7 | 140 | 22322,68 | 22326,10 | +3,4 |
| 36 | 55,30 | 57,18 | +1,9 | 71 | 276,32 | 278,17 | +1,8 | 106 | 898,35 | 932,04 | +33,7 | 141 | 37906,16 | 37910,75 | +4,6 |
| 37 | 59,46 | 58,72 | -0,7 | 72 | 297,09 | 299,72 | +2,6 | 107 | 920,43 | 958,16 | +37,7 | 142 | 37950,75 | 37952,29 | +1,5 |
| 38 | 61,83 | 63,10 | +1,3 | 73 | 300,47 | 307,09 | +6,6 | 108 | 991,75 | 992,43 | +0,7 | 143 | 42641,55 | 42642,89 | +1,3 |
| 39 | 65,73 | 65,99 | +0,3 | 74 | 301,74 | 322,88 | +21,1 | 109 | 1027,49 | 1075,02 | +47,5 | 144 | 42675,93 | 42677,59 | +1,7 |
| 40 | 65,97 | 67,81 | +1,8 | 75 | 313,03 | 327,95 | +14,9 | 110 | 1063,47 | 1104,60 | +41,1 | 145 | 49364,19 | 49366,05 | +1,9 |
| 41 | 71,84 | 72,53 | +0,7 | 76 | 323,58 | 331,86 | +8,3 | 111 | 1077,38 | 1120,68 | +43,3 | 146 | 49400,39 | 49400,58 | +0,2 |

**Table 4.2** – Body Mode Frequency Comparison – With RBB

Lane change differences of the stiffened body vs the non-stiffened body without the RBB (Figure 4.2), show large difference in Y path deviation on the second lane manoeuvre and it is noted that a higher torque is required at the steering wheel to effect the change (+0.2Nm and +0.3Nm respectively), Figure 4.2.

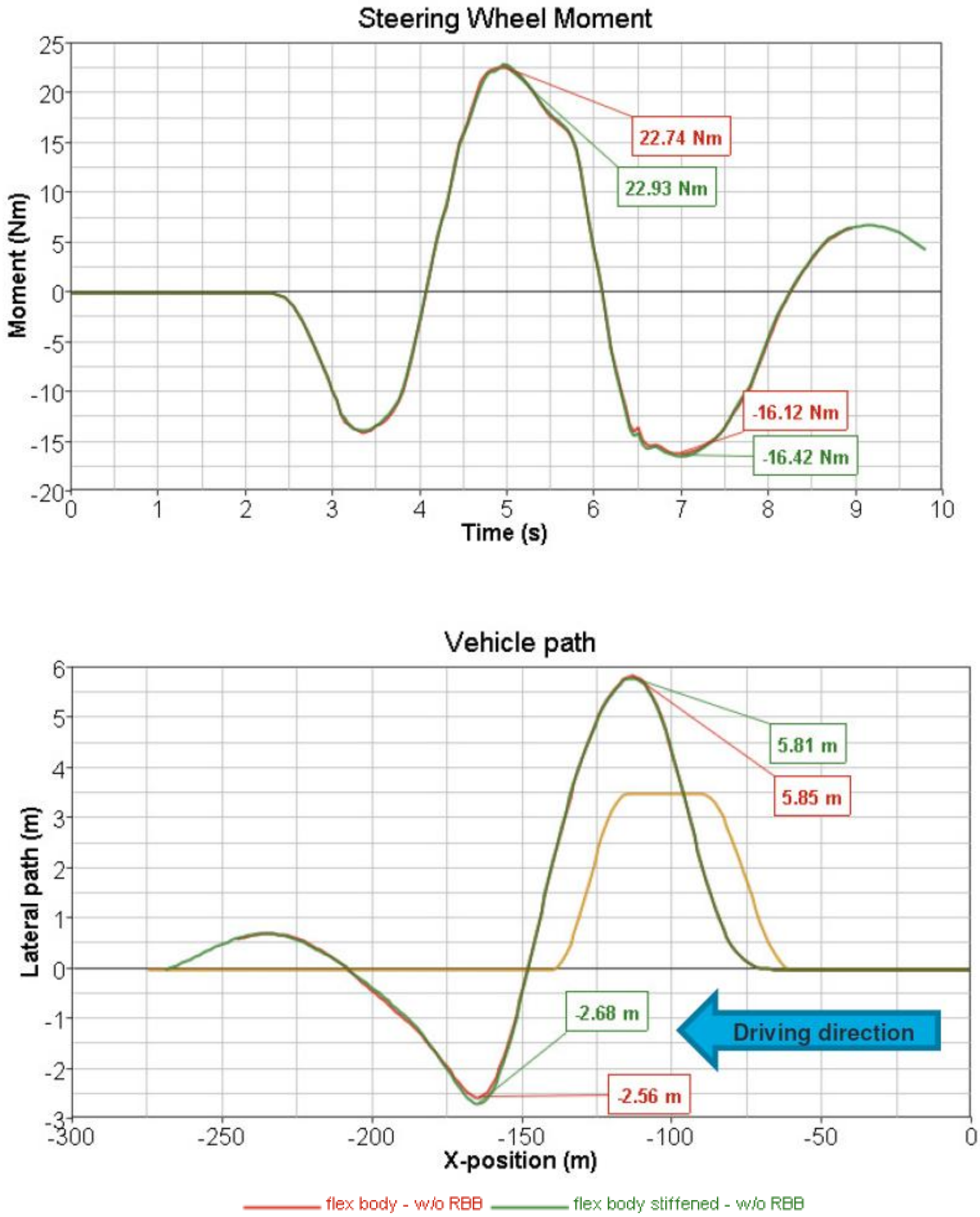



**Figure 4.2** – Body Stiffness Influence on Vehicle Response During Double Lane Change Event without RBB

Higher wheel slip angles were also observed with a stiffened body without RBB, especially during the second manoeuvre. Toe and camber increases were also observed with a stiffened body without RBB, Figure 4.3 and Appendix B.

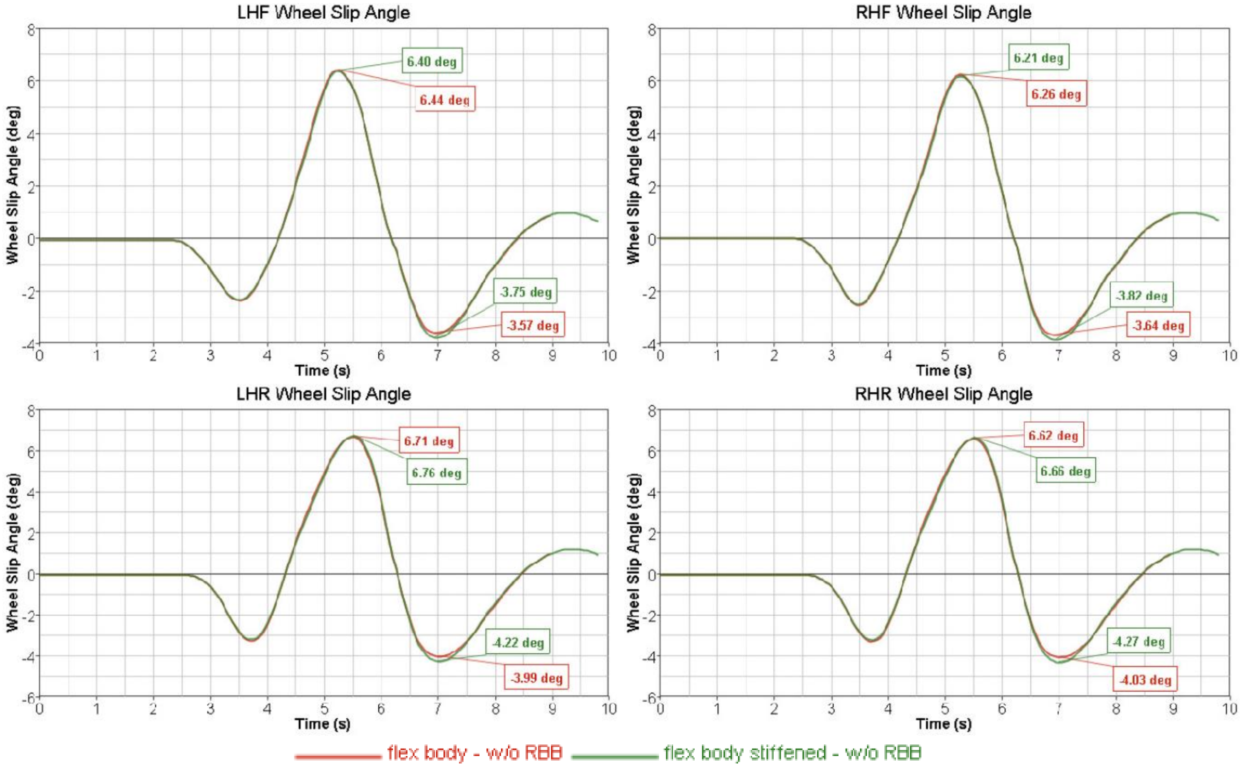



**Figure 4.3** – Body Stiffness Influence on Wheel Slip Angle without RBB

The analysis with the ADAMS model did not however show any significant differences in lane change manoeuvre with a stiffened vs non-stiffened body with the RBB fitted despite the difference in body modes (Table 4.2). The double lane change path variation is insignificant comparing the two bodies, Figure 4.4.

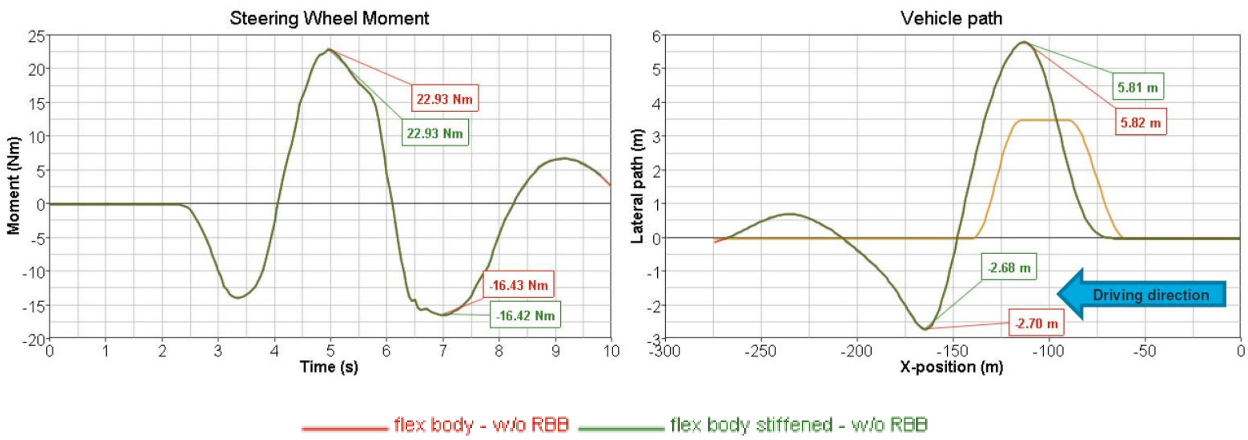



**Figure 4.4** – Body Stiffness Influence on Lane Change Manoeuvre with RBB

These results show that the presence of the RBB when the body is stiffened does not change its handling characteristics. However, when the RBB is removed from a non-stiffened body, the differences are very noticeable.

However, body stiffness improvements were implemented in the assumption that hysteresis would be improved. This is the phenomena that the body stiffness attribute is proposed to overcome.

When comparing these graphs, it can be shown that the lane change criteria do overlay and that increases in body stiffness (by virtue of eliminating high stress points) do appear to overcome differences in RBB presence, Figure 4.5.

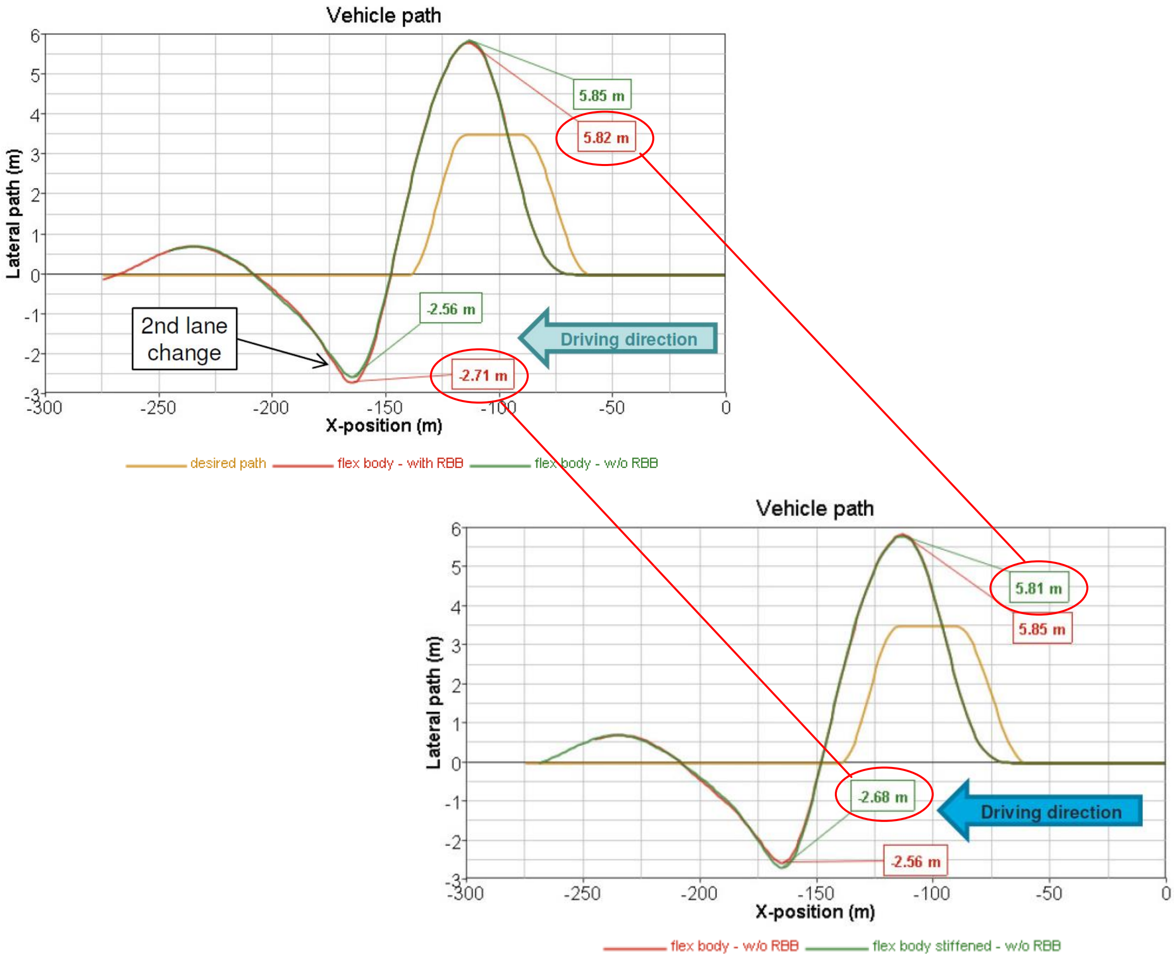



**Figure 4.5** – Lane Change Comparison. Non-Stiffened Body With RBB vs Stiffened Body Without RBB

Wheel slip angles also show similar correlation to lane change effects with the stiffened body without RBB and the non-stiffened body with RBB, Figure 4.6.

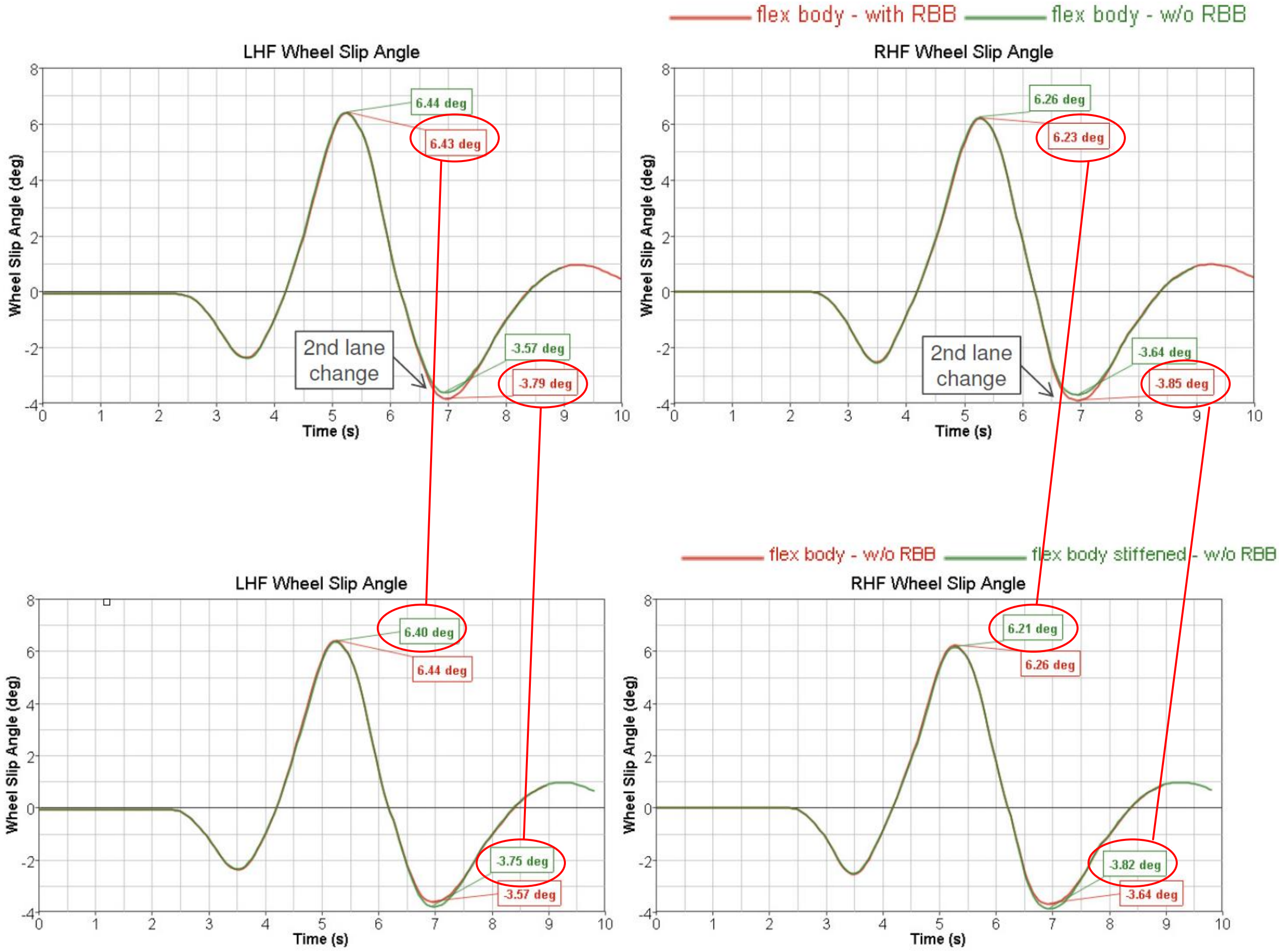



**Figure 4.6** – Wheel Slip Angle Comparison. Non-Stiffened Body With RBB vs Stiffened Body Without RBB

As a final check, the yaw rate and body roll data overlay also indicate good correlation in increased body stiffness; especially in the 2$^{nd}$ lane change manoeuvre, Figure 4.7.

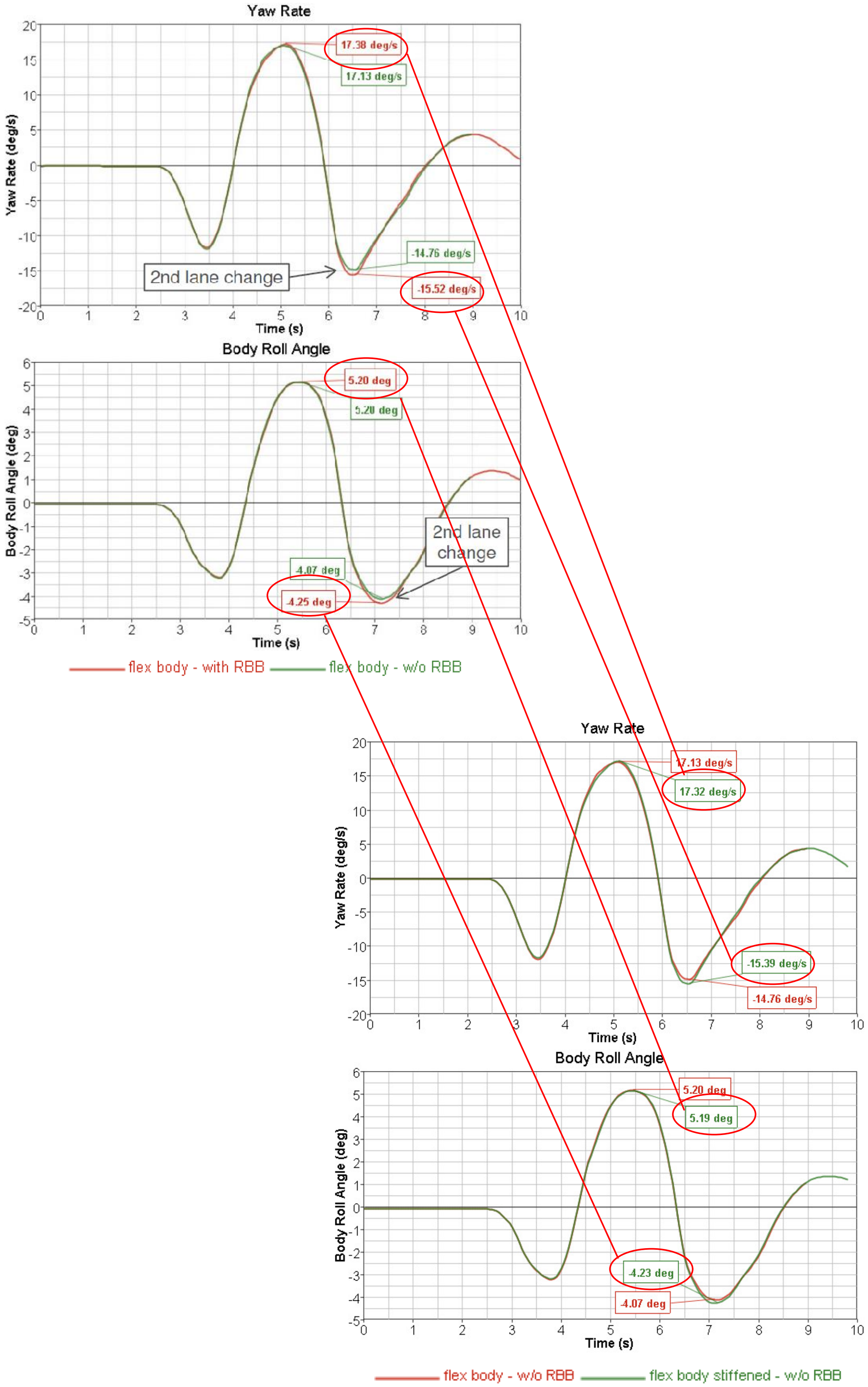



**Figure 4.7** – Yaw Rate and Body Roll Comparison. Non-Stiffened Body With RBB vs Stiffened Body Without RBB

## 4.3 Physical Body Stiffness and Hysteresis Improvements

It must be noted, however, that the deviations in wheel angle, yaw overshoot, yaw rate and body roll are very small. This could just as easily be explained by percentage errors within ADAMS model. To confirm the results, a vehicle was instrumented and tested simulating the ADAMS body stiffness's to demonstrate the RBB contribution to steering behaviour, Figures 4.8, 4.9 and 4.10.

A vehicle was fitted with data acquisition equipment and driven via a robot to remove human response and variation in driver inputs to simulate the ADAMS model. The following inputs were measured:-

- Yaw Rate – yaw (° / sec)
- Lateral acceleration – ay (g)
- Velocity – vx (km/h)
- Roll angle – troll (°)
- Steering wheel angle – swa (°)

The vehicle was driven with and without the RBB to determine the baseline performance of the vehicle as built. The vehicle was also driven by a professional vehicle evaluator to determine the objective steering performance as well as objective data.

The vehicle body was then stiffened in the same areas as the CAE model to determine the impacts to steering improvement based on the Mazda body hysteresis study by Makino, 2004. The vehicle was again driven by a trained evaluator for subjective evaluation.

**Figure 4.8** – Vehicle Configuration for Double Lane Change Test (Outriggers Fitted for Safety)

**Figure 4.9** – Data Acquisition System

**Figure 4.10** – Vehicle Loading Condition (GVM – UDL)

The plots of vehicle response show correlation with the ADAMS model. Albeit the differences are again very small. In order for the vehicle to be robotically driven, outriggers were attached to the front bumper via a solid aluminium crossmember, Figure 4.8, which may have an effect on the vehicle response.

The first plot, Figure 4.11, shows Yaw rate vs steering wheel angle. In order to get the responses on the same graph, scaling factors have been used for pictorial purposes.

Each drive event was done with 3 magnitudes of steering input to get a broad range of responses. Only one graph is shown for reference. All responses are shown in Appendix C.

Reponses with the RBB are labelled ***b*** and labelled ***c*** without RBB (eg. $swa_{\boldsymbol{b}}$ and $swa_{\boldsymbol{c}}$).

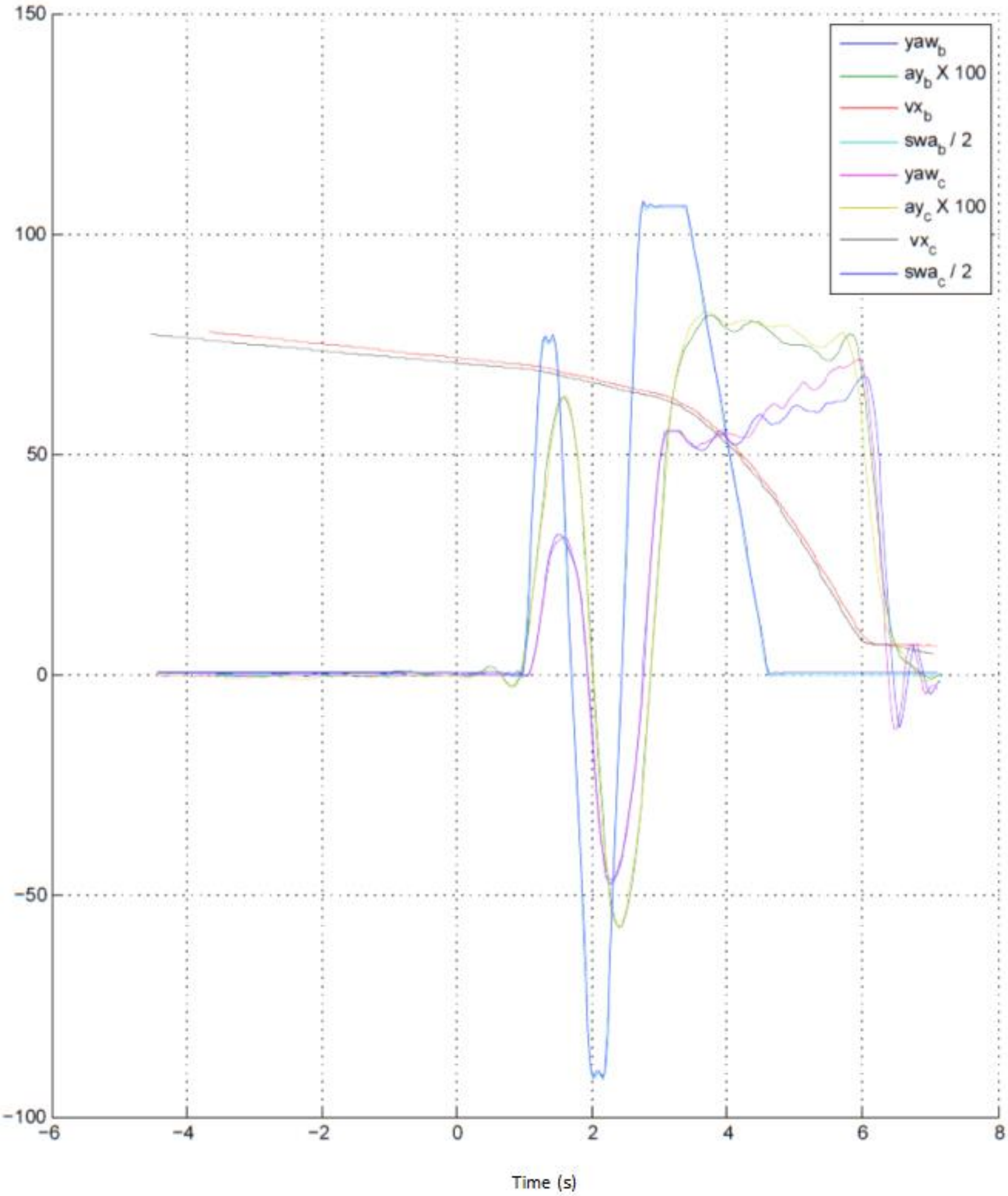



**Figure 4.11** – Vehicle Response of Yaw Rate vs Steering Wheel Angle

Steering wheel angle (swa) and velocity (vx) are ignored as they are constant. The yaw rate (yaw) and lateral acceleration (ay) are the important factors in determining the vehicle responses in this test – Appendix C.

Peak readings of ay are very similar, albeit at marginally different times. As the scales are x100 they can also be ignored.

Yaw rate in this case is the largest difference as this is not influenced by scaling of the graphical data. The difference of yaw rate is the noticeable characteristic of steering response to a driver as this gives the perception and confidence of response.

Inspecting the yaw rate more closely, it can be observed that the response of the vehicle without the RBB had led to a slower response of approx. 5°, Figure 4.12. But this was only noticeable after the double lane change manoeuvre when the vehicle was in the process of stabilising. No measureable differences were noted during the manoeuvre itself.

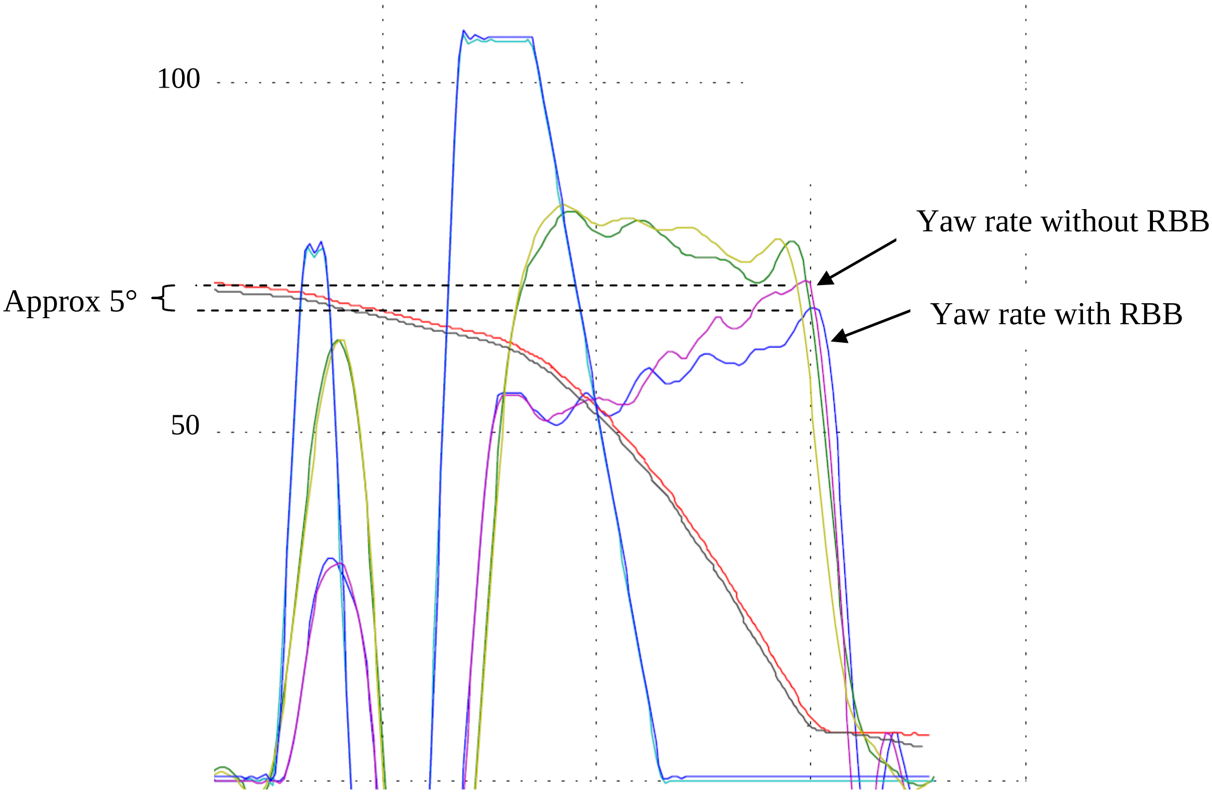



**Figure 4.12** – Yaw Rate Response With and Without RBB

The vehicle roll angle was also measured against swa. The differences in roll are less pronounced than the yaw rate but are still evident. Both vehicle conditions (with and without RBB) show that the vehicle had oscillations after the second lane change manoeuvre.

It can be noted however that the vehicle with RBB has a lower overall magnitude during vehicle stabilisation and is damped quicker than without the RBB – albeit with no measureable difference during the primary event. Roll, along with yaw, also contributes significantly to driver steering perception, Figure 4.13.

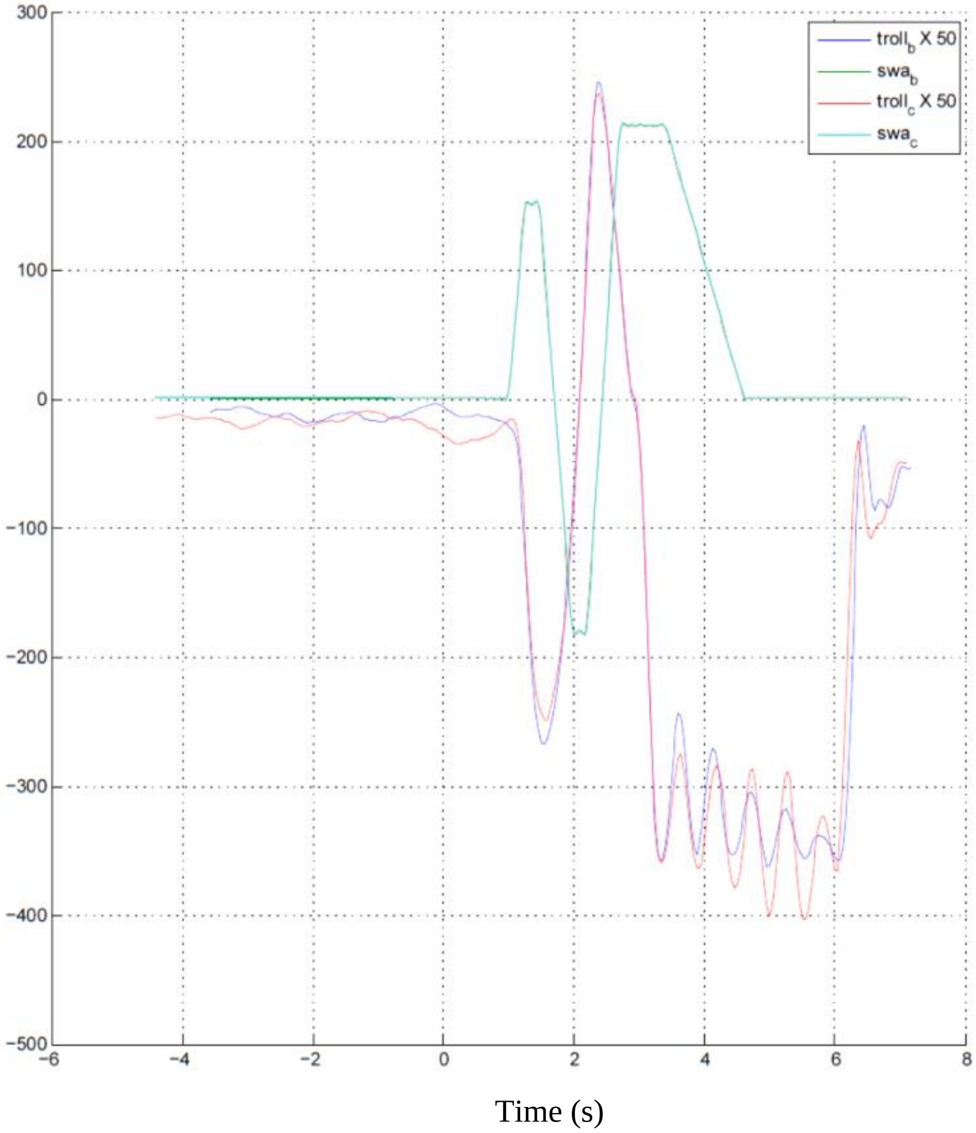



**Figure 4.13** – Vehicle Response of Roll vs Steering Wheel Angle

Neither of these parameters overlay with the ADAMS simulation. It is clear that the variation in the ADAMS model magnitudes is not replicated in the vehicle test, Figure 4.14.  This will be discussed in Chapter 5.

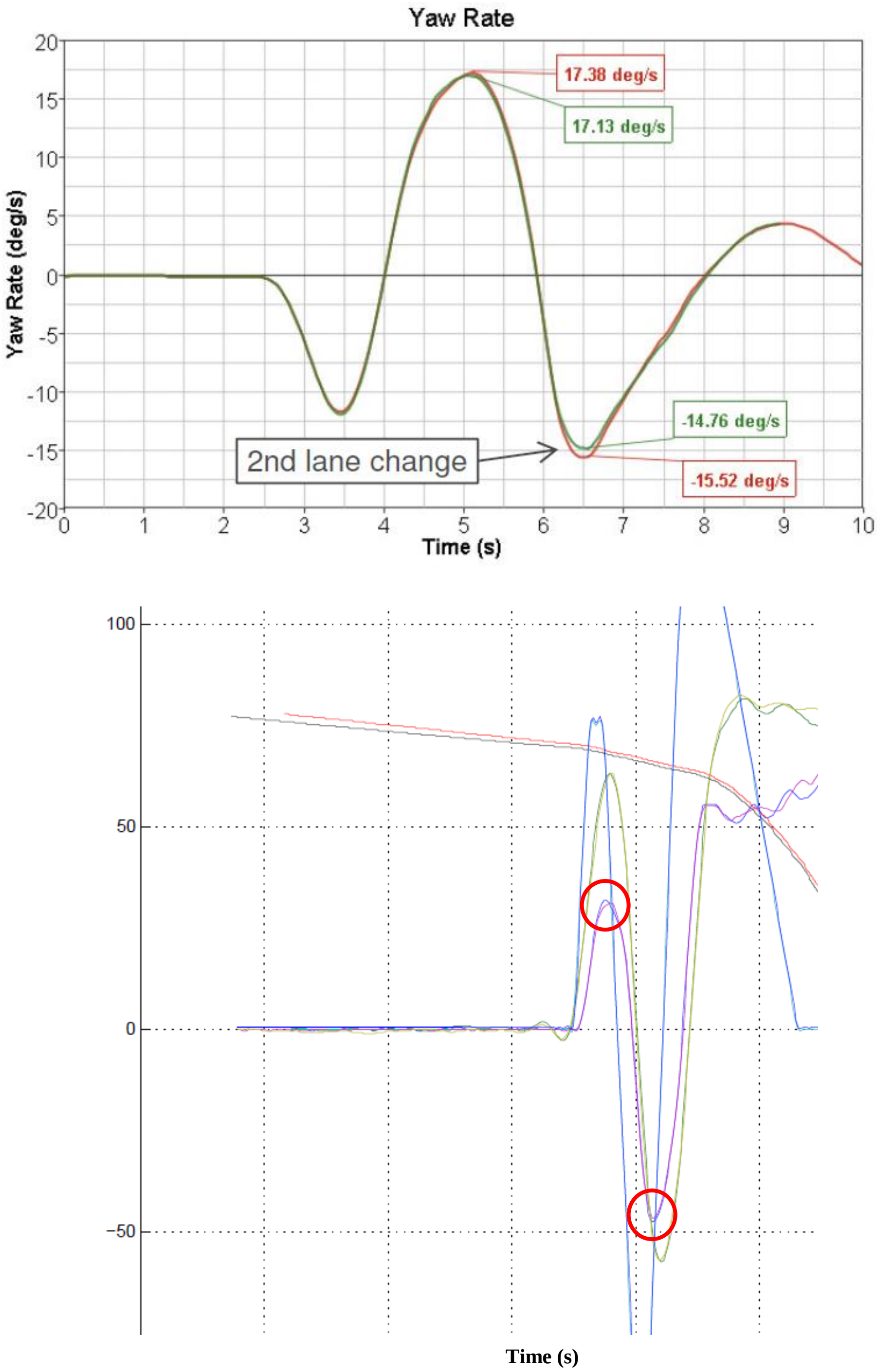



**Figure 4.14** – Yaw Response, Simulation vs Driven Vehicle

The vehicle body was stiffened to simulate the modification in the ADAMS model.  This is shown in Figures 4.15, 4.16 and 4.17.

**Figure 4.15** – Stiffened Roof Bows

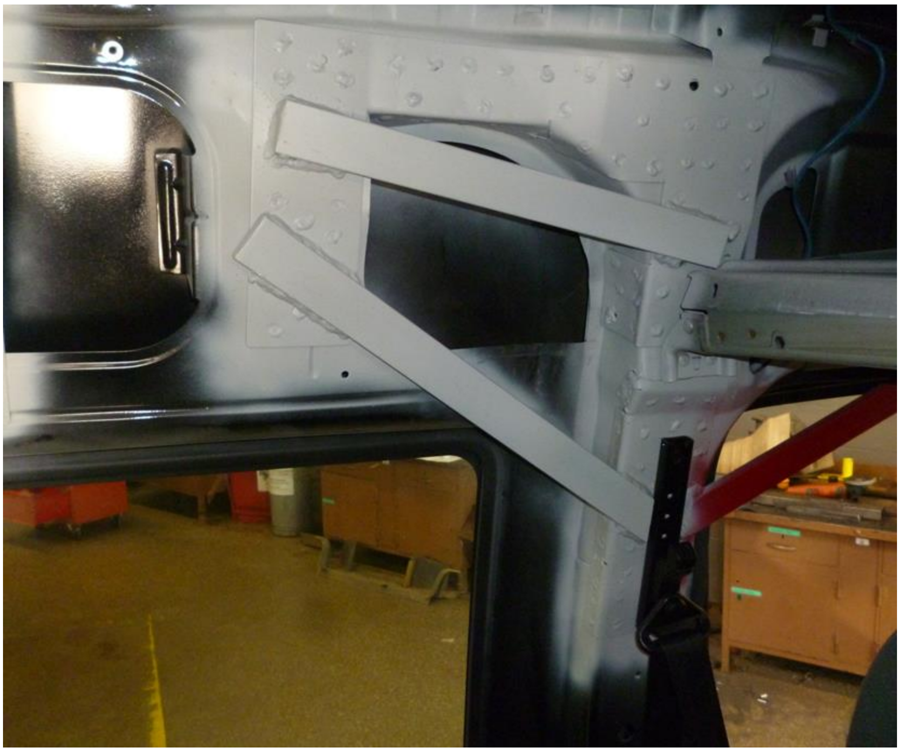


**Figure 4.16** – Stiffened B-Pillar

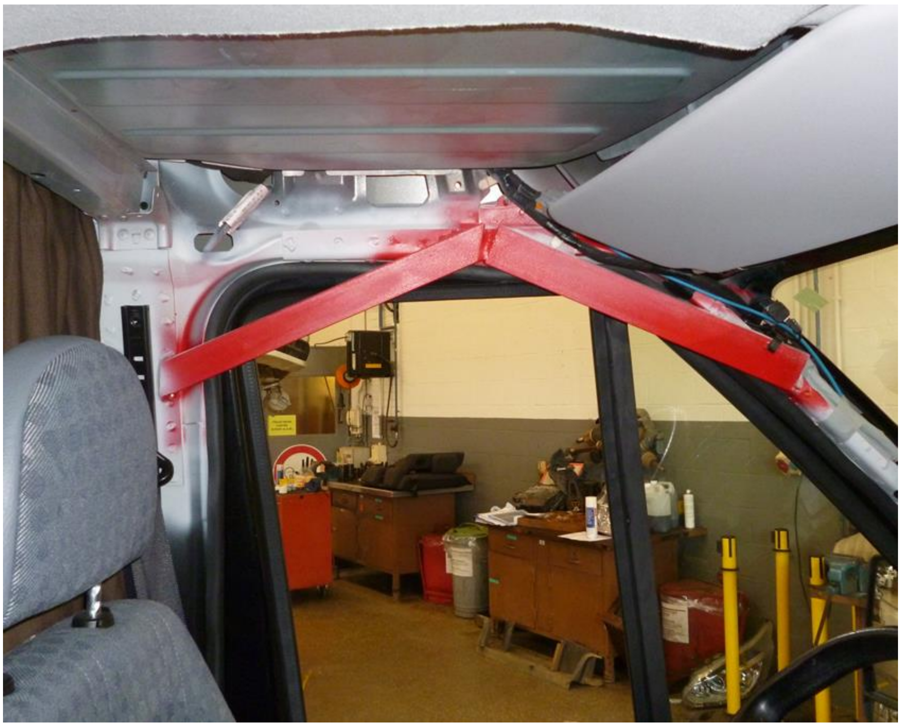


**Figure 4.17** – Stiffened Door Aperture

The vehicle was then driven with the same robot on the same events at identical speeds and conditions.

These also do not show a significant variation of the vehicle yaw or roll response during the double lane change manoeuvre, Figures 4.18 and 4.19. Full results for the stiffened body are shown in Appendix D.

The roll measurements initially show less slip and vehicle reaction to stabilisation than the previous measurements. However, as speeds were increased, the frequency of outrigger contact to the ground became more prominent. This is discussed in detail in Chapter 5.

The overall status of the stiffened vehicle did not show any measureable differences regarding yaw and roll prior to the body being stiffened. Results from stiffened vehicle are denoted ***d*** (ie. $yaw_{d}$).

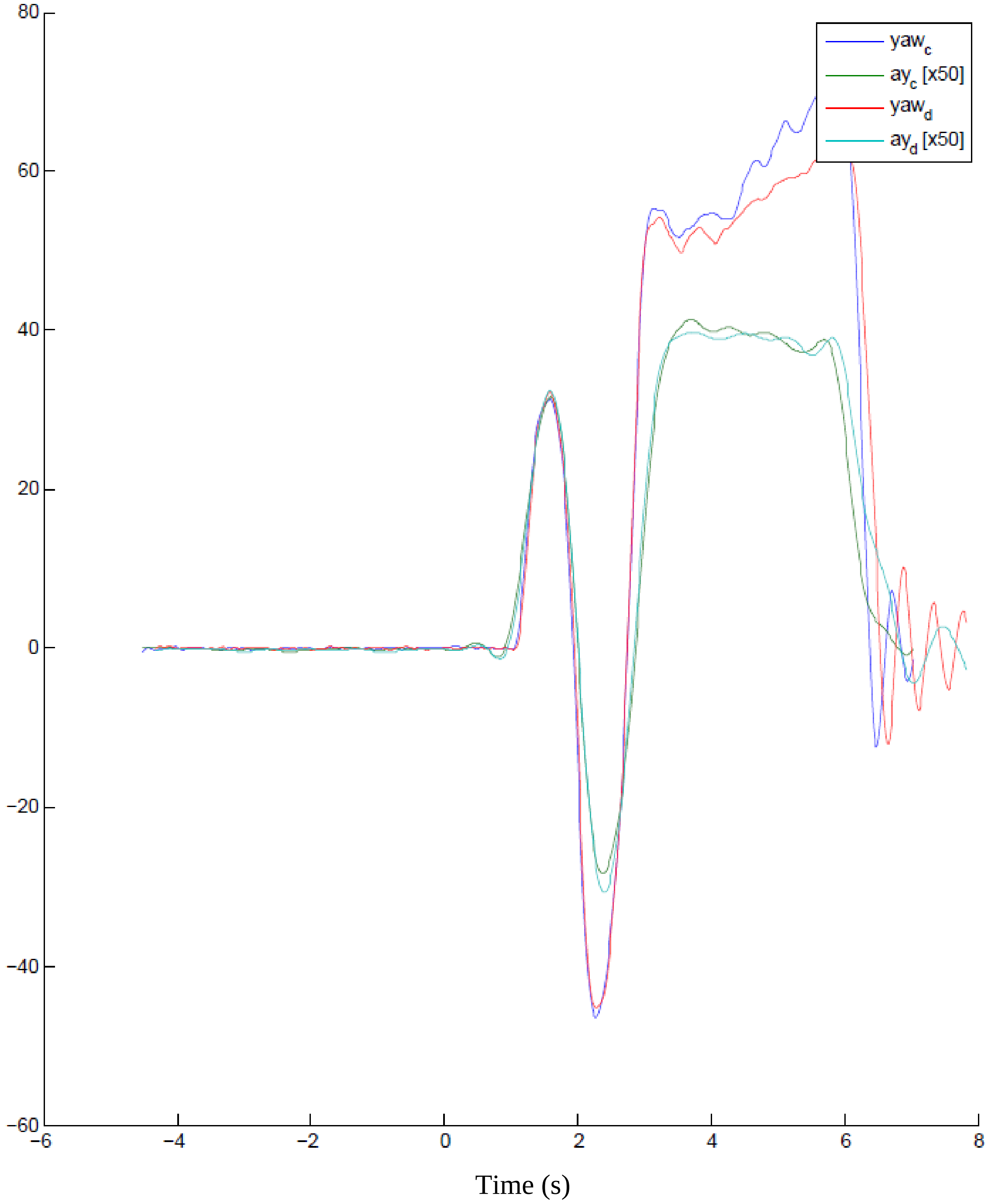



**Figure 4.18** – Vehicle Response of Yaw Rate vs Steering Wheel Angle – Stiffened Body

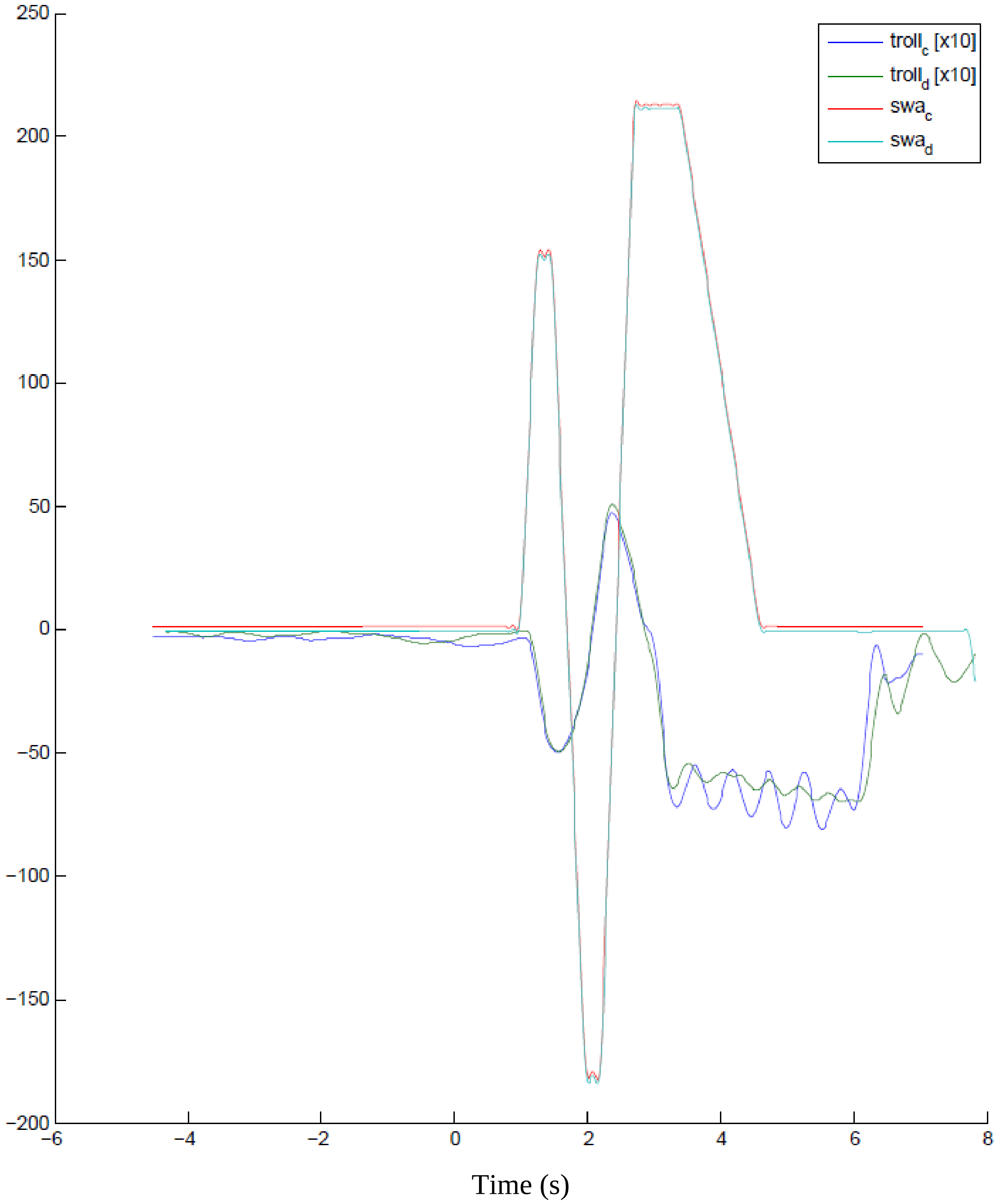



**Figure 4.19** – Vehicle Response of Roll vs Steering Wheel Angle – Stiffened Body

From this data, it is inconclusive to determine the actual effects of the body stiffening actions in comparison to the ADAMS model. However, this will be discussed in detail in Chapter 5.

## 4.4 Subjective Vehicle Behaviour

Vehicle dynamicists from Lommel Proving Ground in Belgium were asked to subjectively drive the vehicle with and without RBB in both the original and stiffened condition to rate the level of improvement.

The full report is shown in Appendix E.

**Vehicle evaluation as received**

01/07/2014 Config: GVM-UDL
14u, Sunny Weights:

| | | |
|---|---|---|
| 785 | ↑ | 753 |
| 923 | | 902 |

kg

Track: 36°C
Amb:18.5°C

Without bumper beam (vs with)

Steering: On center feel gone, no torque build up no V-shape. Delayed and very progressive response. -1.0 VER

Handling: No confidence feel due to poor steering. Roll control worse, less controllable lane change -1.2VER. Borderline acceptable for Trustmark.

**Vehicle evaluation with reinforcements**

06/11/2014 Config: GVM-UDL
16u, Clouded Weights:

| | | |
|---|---|---|
| 783 | ↑ | 752 |
| 924 | | 908 |

kg

Track: 10°C
Amb:8.5°C

Without bumper beam (vs with)

Steering: Slightly bigger dead window on center, a bit more torque build up (V-shape) on center. Similar response off center. -0.2 VER

Handling: Similar performance, no real difference noticable.

**Conclusion**

**Removing the rear bumper beam on the vehicle results in an onacceptable drop of VeD performance (Trustmark borderline). When removing the bumper beam on the reinforced vehicle, the drop in VeD performance is almost negligible (-0.2VER for steering)**

**Figure 4.20** – Subjective Handling Assessment

This report clearly shows that the body modifications and stiffness / hysteresis improvements are noticeable and correlate with the ADAMS model.

## 4.5 Summary

From the ADAMS model analysis, there appears to be strong correlation between body stiffness and RBB presence. This indicates that there is good correlation between body hysteresis and body stiffness perception and that this is quantifiable (as suggested by Makino, 2004).

The objective physical vehicle testing did not show the same correlation and suggests that the ADAMS results were software / analytical rounding errors appearing to contradict the hysteresis theory.

The subjective vehicle testing however, did show the correlation to the ADAMS model and proved that body stiffness / hysteresis improvements are the key to delivering the RBB removal.

Conclusions and recommendations will expand on these further.

# 5. Results, Conclusions and Recommendations

## 5.1 Summary of Results

In empirical testing and driving, it has been historically difficult to understand why the RBB deletion affects vehicle steering behaviour. Many attempts to simulate the RBB via lower cost solutions on the vehicle (simulating weight, stiffness or a combination of the two) have not been successful.

Analytical methods have also been unable to predict the steering degredation for two main reasons:-

- The traditional ADAMS chassis model for the suspension and steering does not include the full flexible body and only assumes rigid body utilising corner weights
- The flexible body model doesn't take the ADAMS suspension inputs into account during FEA analysis at all

The results of the investigation into marrying these hitherto separate models into one parametric model, has enabled the steering effects of body stiffness to be analysed.

This model was able to demonstrate a difference in steering input during a specific double lane change event which also correlated with measured understeer gradient measurements from a physical vehicle.

This model was then able to be used to determine how the steering attribute could be maintained whilst realising the desired deletion of the RBB.

The investigation into body hysteresis and stiffness was then analysed to determine if the study did indeed have the effects suggested (Makino, 2004).

Physical testing to back up this data did not confirm the effect of body stiffness on the steering perception. There are issues with the vehicle testing that need to be considered, however, before any conclusions regarding the physical testing can be drawn.

## 5.2 Results Analysis

The analytical model results are conclusive, if not overwhelmingly so. The physical test results however, did not show any improvement. There could be a number of reasons for this:-

- The stabilisers mounted to the vehicle
- The weight added to the vehicle to simulate GVM
- Method of stiffening actions made to the vehicle body

### 5.2.1 Stabilisers Mounted to the Body

The stabilisers mounted to the body are required during the robotic driving ensure the vehicle does not roll over. However, they are mounted to the vehicle in such a way as to potentially simulate the RBB. This could account for the apparent lack of measurable difference with or without the RBB in either unstiffened or stiffened condition, Figure 4.8.

### 5.2.2 Weight Cages Added to Vehicle

The weight cages added to the vehicle are standard items added to ensure UDL in the load area. They are however tied down and linked together to the load floor to prevent movement or shift during handling manoeuvres.

As in the previous case, this could inadvertently add structural stiffness back into the vehicle body to simulate the stiffness effects of the RBB.

### 5.2.3 Body Stiffening Actions

The actions taken to the body to simulate the ADAMS stiffness increases also increased the weight of the vehicle. It was noted that during the second tests after the stiffening actions that the vehicle was approx. 70kg heavier. Weight blocks had to be removed from the cages to keep the same axle loads.

Stiffening action on the vehicle body in the ADAMS model can be done without compromising the weight assumptions and loading behaviour.

In reality this means that although the vehicle weights are the same, the centre of gravity of the vehicle was increased which would have induced a lot more roll. This borne out by the testing team who noted an increase in the amount of contact between the stabilisers and the road contact at high amplitudes compared to the unstiffened vehicle.

Figure 5.1 shows the differences between the stiffened vs unstiffened body at the highest steering wheel angle amplitudes.

The plateaus at the top of the curves are evidence that the stabilisers hit the ground.

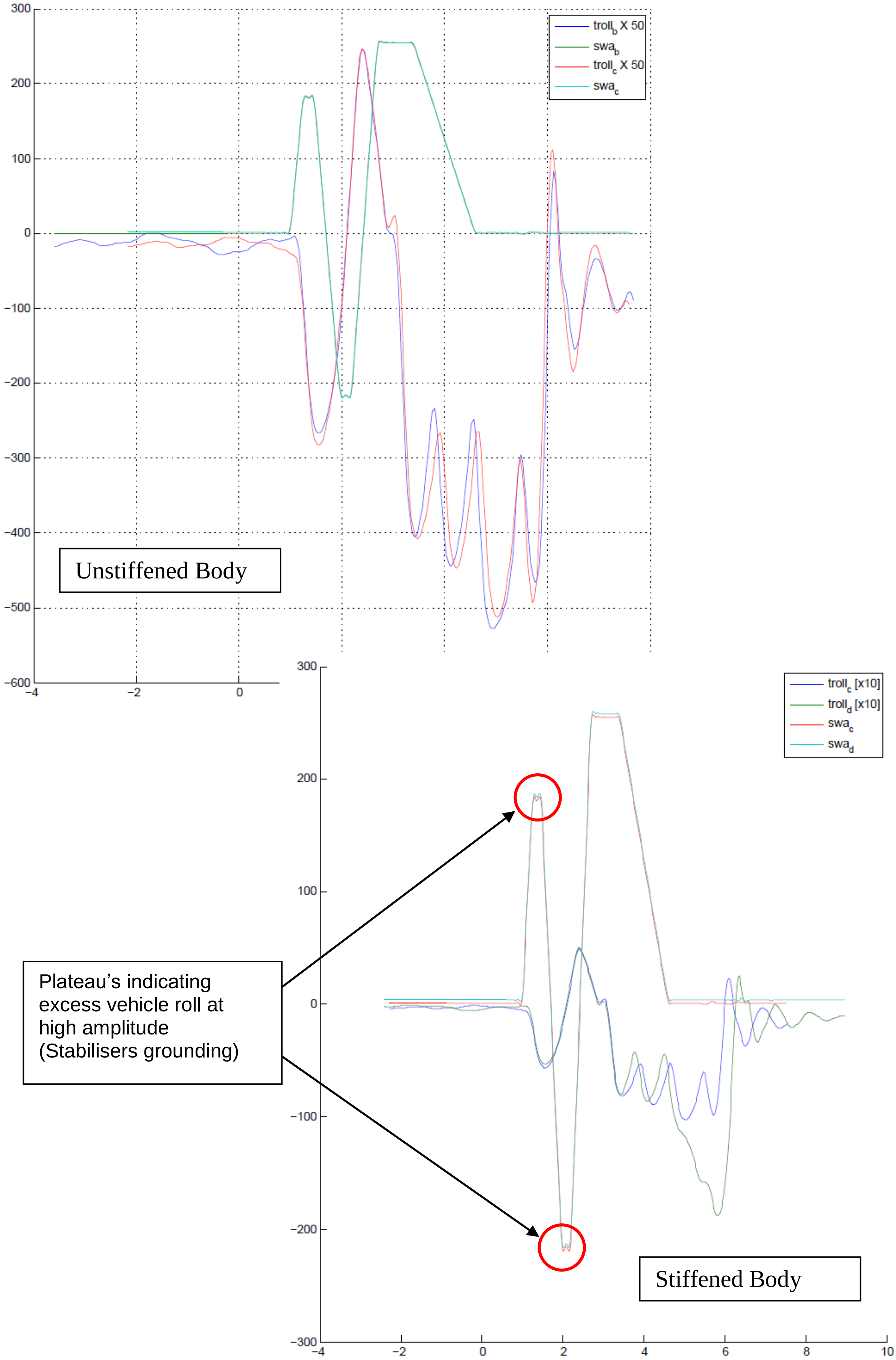



**Figure 5.1** – Roll Response at High Steering Wheel Amplitudes

At the same time it should be noted that traditionally, higher CoG's and roll rates (caused by higher CoG) would be detrimental to vehicle response during lane change manoeuvres. In this case however, the differences are negligible and so it could be argued that the stiffening actions have indeed had some effect – even if the vehicle testing cannot be relied upon 100%.

## 5.3 Conclusions

The determination to the effects of body hysteresis rather than pure stiffness did conclude that the steering attribute could be maintained with the deletion of the RBB analytically.

This is important because in the current manner of analytical methods with separate body and chassis models, the effects of one on the other are ignored. As such the body engineering team concentrate on achieving a stiffness target (as stiff as is reasonably possible) and the chassis team develop the ride and handling attributes assuming the body to be infinitely stiff. In reality, neither of these two methods predicts a rounded or complete status of the vehicle behaviour.

Makino's suggestion that high body stiffness, whilst in itself is desirable, does not correlate to customer perception of rigidity, was the key to determining the best method to quantify this.

The conclusions from this thesis have determined that total vehicle modelling methods currently used, cannot establish or predict the complete vehicle ride and handling status.

Instead, the current method of analytical research focuses solely on the ‘ideal’ ride and handling model in ADAMS and relies on vehicle dynamicists to fine tune the vehicle empirically.

As this is the standard method of vehicle development, the opportunity to change must be considered.

The gestation period of vehicle body development is very long although the tuning window during this period for stiffness improvements is very short. It is therefore important to understand and quantify the requirement for body hysteresis at an early stage.

Changes to BIW design are very expensive so having a detailed understanding of the mechanics of the total vehicle is essential to avoid expensive churn in the developmental process.

However, this report has shown that the total vehicle modelling process can work and can be used to establish analytical sign-off both before and after the vehicle is launched.

Ultimately, targets for body hysteresis should be established, which the engineering team could achieve holistically, would be added to the development statement of each program. Extra emphasis could then be given to known areas of body weakness in CAE without the requirement for expensive vehicle prototypes and testing.

As could be seen from the analysis in section 4.2, the base vehicle responses of the vehicle with RBB can be re-established by reducing high stress areas whilst eliminating the RBB.

It should be noted however that the vehicle response with the stiffened body with the RBB in situ, didn't have a significant influence in the double lane change steering response, Figure 5.2 & Appendix B.

It can therefore be concluded that the desired hysteresis can reach a finite point and that further improvements will not add value when determined analytically.

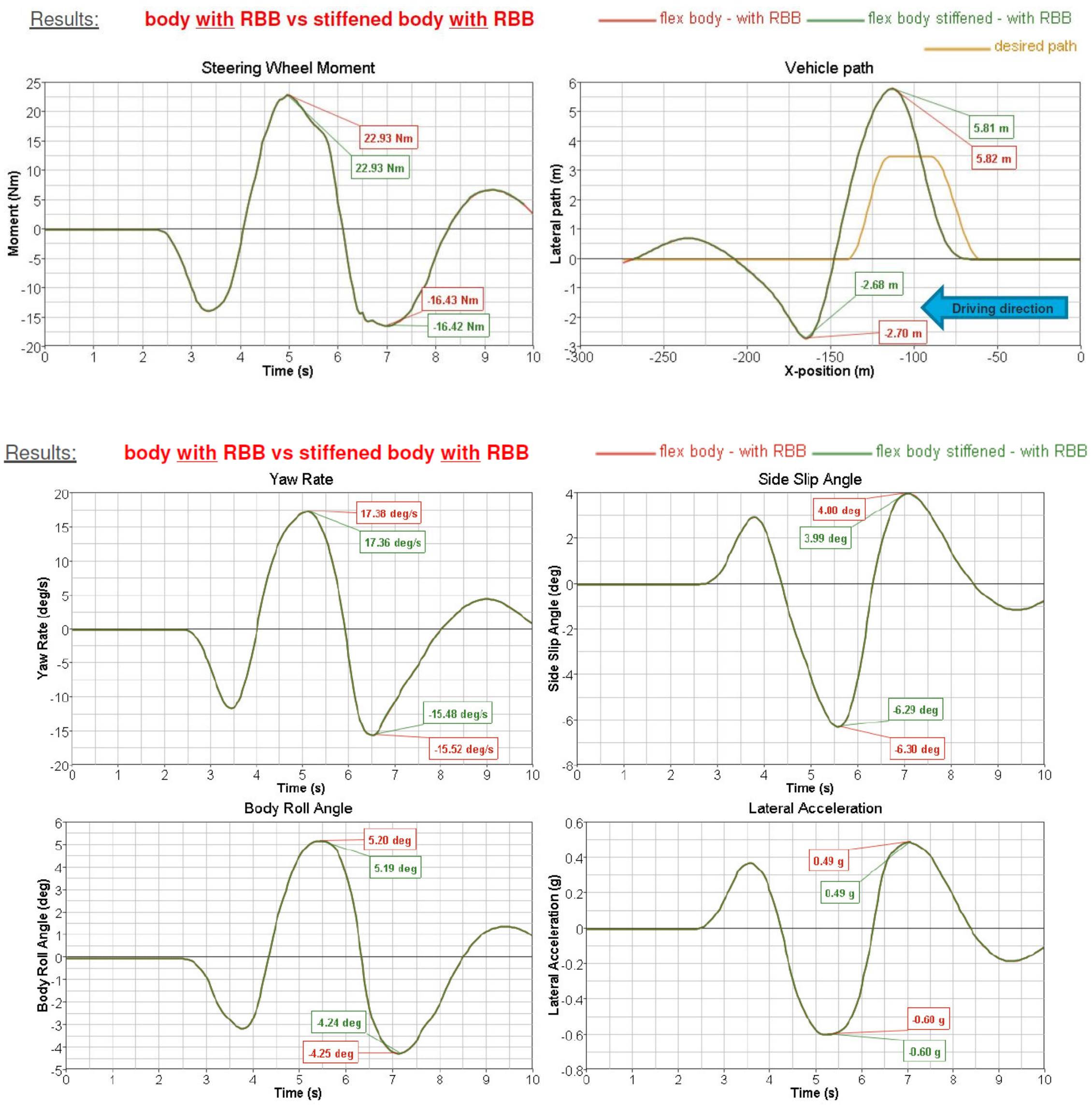



**Figure 5.2** – Vehicle Response Comparison. Stiffened Body With RBB

These responses show insignificant differences in any of the measured responses during the double lane change manoeuvre. This leads to the conclusion that the RBB was artificially adding body stiffness to the structure which is secondary to its' intended function. In the case of the vehicle as designed today, this means that it cannot be removed without deteriorating the vehicle steering perception and transient lane change behaviour.

The analysis with the physical vehicle has inconsistent and inconclusive results which cannot be relied upon for accurate results. Further work on physical vehicles is required to fully determine where and how stiffness is improved.

## 5.4 Recommendations and Further Work

As discussed in section 5.3, there is a limited period during BIW development to eliminate high stress areas. This time should therefore be baked into the development timing and as early in the gestation period as possible.

As well as this, it is recommended that a full vehicle analytical model should be developed in ADAMS Chassis, as described in section 3.7. This will have the benefits of aiding the development team prior to launch as well as being infinitely variable to be used for post launch development and for next generation vehicle target setting and across all vehicle models.

As has been shown, the correlation between body stiffness and hysteresis, suggested by Makino, does appear to be influential to the perceived rigidity feel. Furthermore, it has been shown using the full vehicle analytical model that this perception is also real and quantifiable.

To establish effectiveness of the full vehicle model, it is recommended that the attribute differences identified be verified on working vehicles to establish true correlation.

Although the model does show that there are measurable differences, it is not possible to quantify the objective and subjective differences until a working model can be built.  This work plan should consist of:-

- Measuring a production vehicle to determine the real world hysteresis
- Drive evaluations of standard vehicles with and without RBB
- Modify the vehicle to reduce the traditional high stress areas without affecting CoG or roll response
- Re-measure the hysteresis
- Re-drive the vehicle to determine improvements in steering perception

Although the full vehicle analytical model does show significant differences in response during the double lane change event, there are a lot more events that can be simulated in ADAMS.  These should be re-run to determine the major response differences during the entire suite of proving ground events involving transient steering response.

Once confidence in the ADAMS model correlation is established, the attributes can be established and monitored entirely in the ADAMS model.

As stated in section 1.4, the steering feel deterioration caused by the RBB deletion affects all types of vehicles.  In the interests of completeness, this study should be conducted on a range of different vehicles to assess the full effects of a parametric full vehicle ADAMS model.

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# Body Structure and Vehicle Dynamics at Mazda

Presented to Ride and Handling Forum
29July2004 by David Russell

Mazda Proprietary Information presented by David Russell DRUSSE10
Mazda contact: Kimiaki Makino, VehDyn General Manager

# Caveats

- I did not create this information.
- I will present it the way I understood it.
- I will collect questions that I cannot answer and share Mazda's responses with the Forum

Mazda Proprietary Information presented by David Russell DRUSSE10
Mazda contact: Kimiaki Makino, VehDyn General Manager

# Background

Mazda treats body structure as a Vehicle Dynamics tuneable early in the program with Vehicle Dynamics CAE and Development providing targets and design input into the Body Structure. This is a small but important window of time to adjust the design.

Both CAE and mechanical modifications to early prototypes are used to drive the Body Structure design. Build a solid foundation for the future suspension tuning.

Mazda Proprietary Information presented by David Russell DRUSSE10
Mazda contact: Kimiaki Makino, VehDyn General Manager

# Body Rigid Feel

"Body Rigid Feel" is a subjective category that Mazda rates subjectively during development and sign-off. Mazda believes this category is very important to the delivery of quality feel to the customer.

It is composed of both Ride and Steering responses of the body.

- Does the body itself feel solid?
- Is the rigidity uniformly distributed or can you feel lags and/or aftermotion in the body?

Mazda Proprietary Information presented by David Russell DRUSSE10
Mazda contact: Kimiaki Makino, VehDyn General Manager

## How Mazda Designs for Body Rigid Feel

Body Structure targets from a Strg/Hdlg standpoint are global stiffness and stiffness balancing from front to rear of the vehicle.

- Global stiffness is described as static torsional and bending stiffness.
- Distribution balance to be shown as a diagonal displacement of key points (Upper A to Lower D pillar, for example).



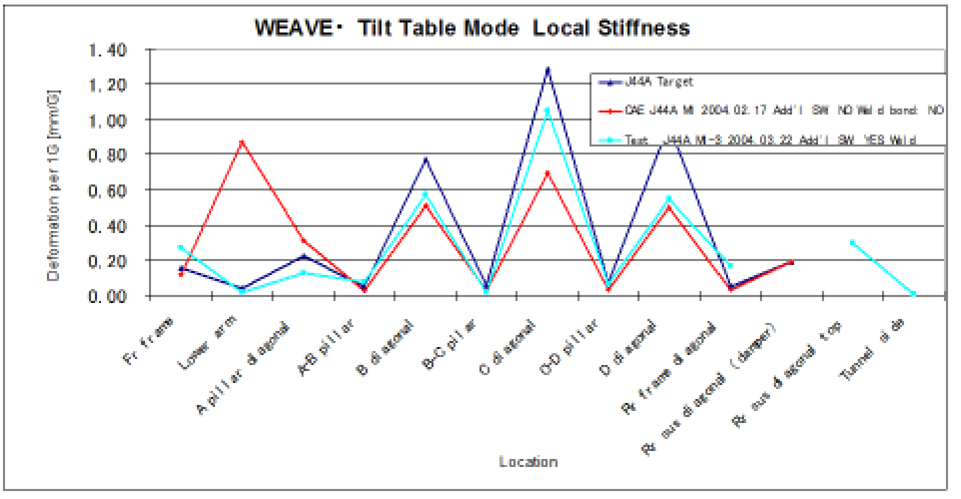
## How Mazda Designs for Body Rigid Feel

Mazda uses a Tilt Table to measure body response to steering and a 4post Rig to measure twist during long wavelength ride events to capture the subjective "rigid feel"

Mazda selects targets for each event for both Overall Stiffnesses ***and*** Distribution of Stiffness

Mazda Proprietary Information presented by David Russell DRUSSE10
Mazda contact: Kimiaki Makino, VehDyn General Manager

# How Mazda Designs for Body Rigid Feel

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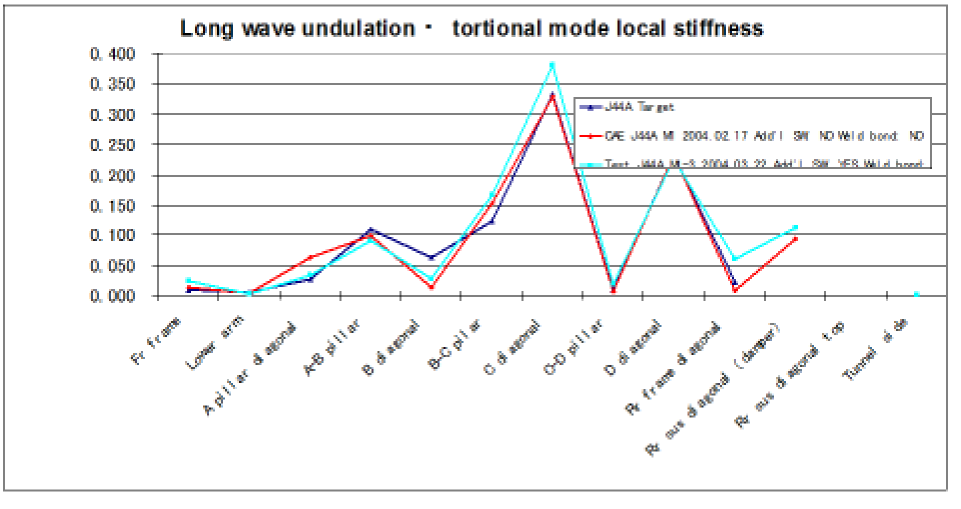
# How Mazda Designs for Body Rigid Feel




Mazda Proprietary Information presented by David Russell DRUSSE10
Mazda contact: Kimiaki Makino, VehDyn General Manager

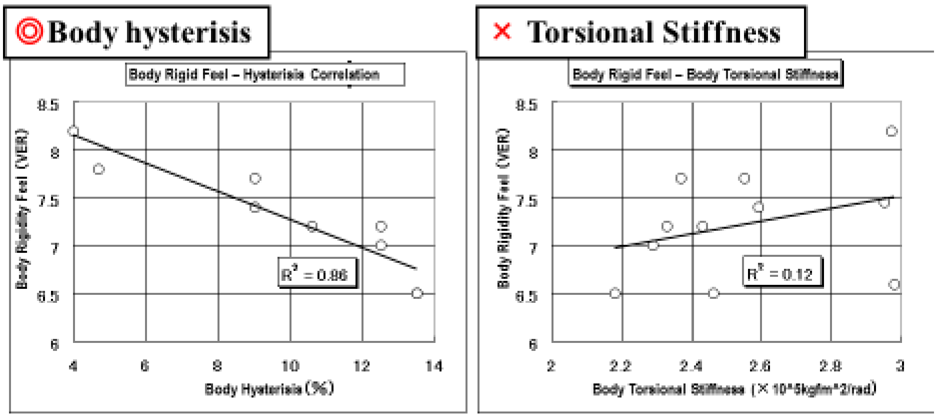
# How Mazda Designs for Body Rigid Feel

Mazda has recently developed a new metric
that contributes to Body Rigid Feel →
Body Hysterisis



## Hysterisis versus Global Torsional Stiffness

**Body Rigid Feel is highly correlated with Body Hysterisis, NOT Torsional Stiffness**

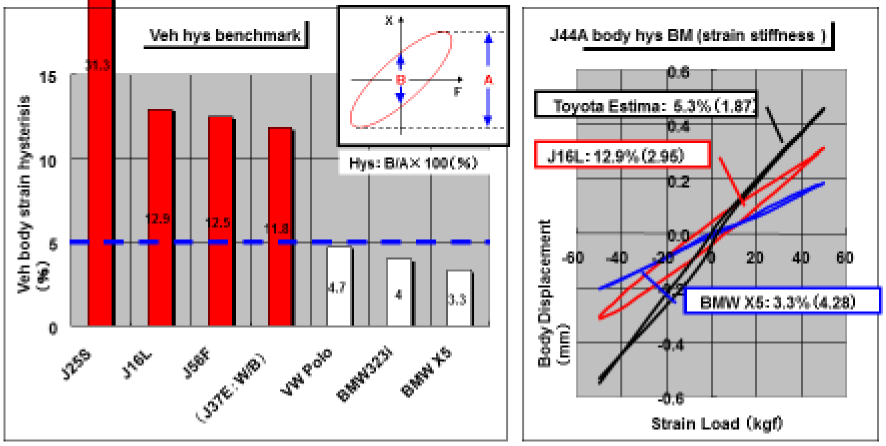



**Note: VW Polo has hysterisis of 4.7% but is soft torsionally at 1.69 (x10e5kgfm^2/rad)**

Mazda Proprietary Information presented by David Russell DRUSSE10
Mazda contact: Kimiaki Makino, VehDyn General Manager

## How does Mazda define Body Hysterisis?

**European Steering/Handling Leaders have small body hysterisis. Mazda believes that <5% should be their future target.**



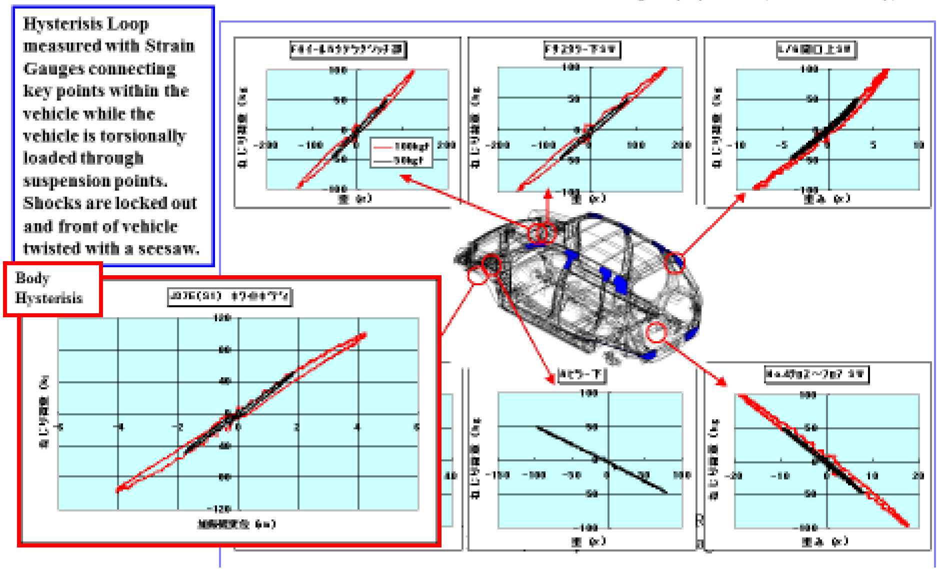

Mazda Proprietary Information presented by David Russell DRUSSE10
Mazda contact: Kimiaki Makino, VehDyn General Manager

## 2.0 Root Cause and Reduction Method for Body Hysterisis

1) Hysterisis loop at SpotWelds and notches where stress concentrate create body hysterisis.

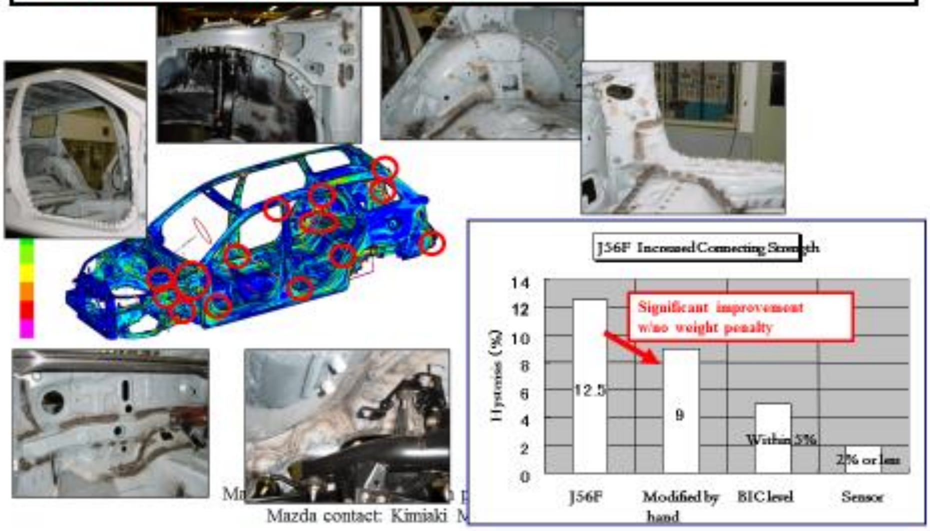
2) Hysterisis band increases as stress amplitude increases. Areas of low stress show no hysterisis.

→ Reduction in stress at SW and notches where stress concentrate is effective in reducing body hysterisis. (Same as durability)



## 3.0 Effect of Increasing Connection Strength (J56F Mazda 6

Increased strength at connections reduced hysterisis 30%
(Objective: Add SpotWelds at the areas with >10Mpa. Add Electric Welds (equivalent to 1000 spots, approx 50m in length) → Connection strength increase is weight effective.
※Stg/Hdlg and body rigidity feel increased by 0.5 VER.

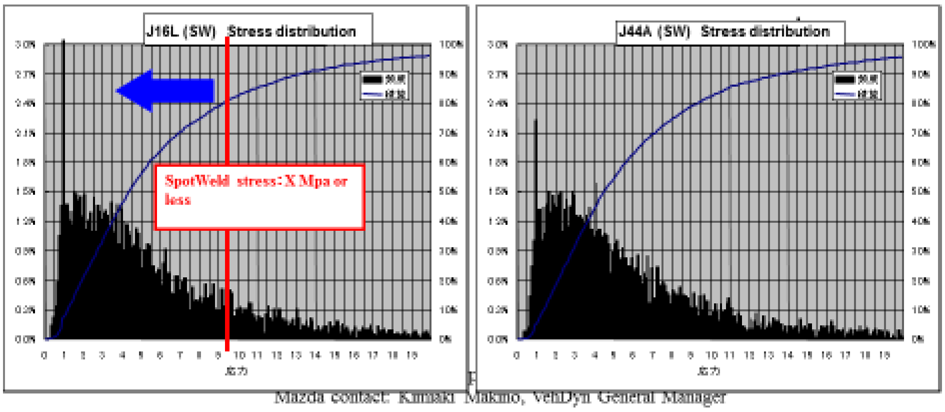



## 4. Body Hysterisis Development Method (J44A M1 Example 1)

How to Predict this improvement with CAE? → Stress levels trend with hysterisis, spreading out stress more uniformally reduces it

CAE used to measure Stress on body SpotWelds with a body static torsional load applied (100kgfm)
Areas demonstrating SpotWeld stress >20Mpa or more were reinforced in a prototype vehicle (approximately 5% of all SW, approx 150 elements effected)

J44A Development



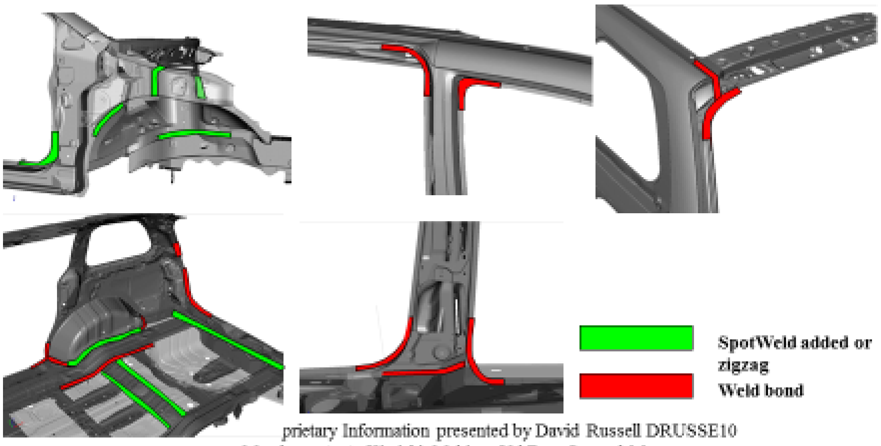

Mazda contact: Kimiaki Makino, VehDyn General Manager

## 4. Body Hysterisis Development Method（J44A M1 Example2）

### J44A M1 Proto Actions

**Proposed body modifications improved body rigidity feel by 0.5 VER (VD team comment)**

Approx 7m of additional welds




prietary Information presented by David Russell DRUSSE10
Mazda contact: Kimiaki Makino, VehDyn General Manager

## 5. Concept for Improving “Body Rigidity”

**1) Body rigidity cascade**

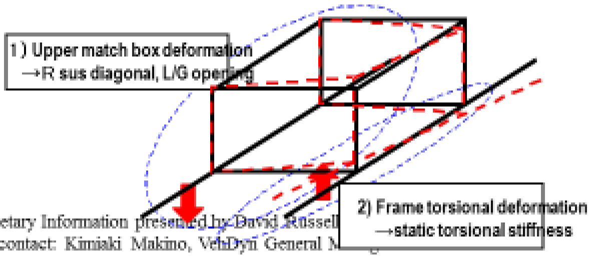
| Item | Body Charcteristics Targ | Analysis Target | Related Items |
|---|---|---|---|
| **Body rigid feel** (x dynamic performance) ※Secondary contribution for VD | **Body hysterisi** (x stiffness) | **Stress** (× stiffness) | Junction Structure (# of SW, weld bond |
| | | | Thickness |
| | | | Shape |
| | | | Stiffness |

**How to set Torsional Stiffness / Hysterisis Target?**

**1) Ensure basic minimum stiffness achieved (Cost of entry, want to use the average of benchmarking)**

**2) Drive hysterisis down to <5% through structure modifications to reduce stress as surrogate for hysterisis**

Example: VW Polo is considered very soft torsionally but has low hysterisis and therefore good Body Rigid Feel. Why pay for Stiffness if can engineer in reduced hysterisis instead?




Mazda Proprietary Information presented by David Russell
Mazda contact: Kimiaki Makino, VehDyn General M

Cologne, October 23rd, 2013

Hereby I, Jérémy Couval, confirm that the CAE investigation of a Transit Adams model driving a double lane change using a flexible body with and without a rear bumper beam has been commissioned by Mr Alan Banks as part of his Master thesis. This investigation and the report are not part of another project or thesis and will not be published in connection with any other study.

Jérémy Couval

Global Chassis Engineering

# V348 DOUBLE LANE CHANGE
# INFLUENCE OF REAR BUMPER BEAM

April 25, 2013

**Jérémy Couval**
Ford of Europe - Chassis CAE
Tel: 70-32988
Location: D-ME5/E5d

Go Further

## AGENDA

Global Chassis Engineering

1. Introduction
2. CAE test setup
3. First analysis: rigid vs flexible body
4. Influence of the Rear Beam Bumper
5. Conclusion & Next steps
6. Additional analysis: body strengthening



Go Further

## 1. INTRODUCTION

Purpose

- Investigate the influence of the Rear Bumper Beam (RBB) on vehicle's behavior

How?

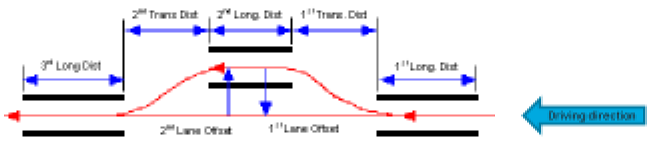
- A-to-B comparison by running double lane change event including car body as a flexible structure in Adams:
  1. with RBB
  2. without RBB
- V347 MWB Low Roof body has been considered

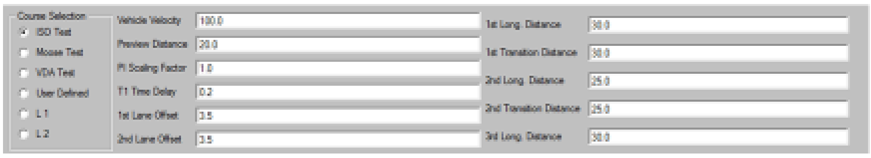



## 2. CAE TEST SETUP (1/2)

Event description

- The test setup is equal to the base Adams/Chassis settings. The Lane change is calculated at a speed of 100 kph. The ideal course is shown in the picture below.



## 2. CAE TEST SETUP (2/2)

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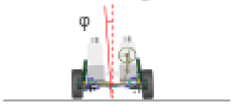
Metrics for results evaluation

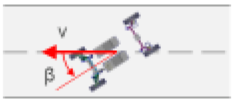
- Vehicle path: Y-position vs X-position
- Steering wheel moment
- Toe, Camber, Caster
- Lateral acceleration

- Yaw rate: describes the angular velocity of the rotation around the z-axis of the vehicle coordinate sytem.

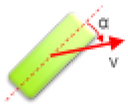

- Roll angle φ: describes the angle around x-axis between the vehicle z-direction and the global z-direction.




- Side slip angle β: describes the angle between the vehicle x-direction and the direction of the vehicle velocity vector.




- Tire side slip angle α: describes the angle between the rolling wheel's velocity vector and the direction towards which it is pointing.






## 3. FIRST ANALYSIS: RIGID VS FLEXIBLE BODY (1/2)

Global Chassis Engineering

- Purpose is to check that flexible body works well and leads to similar behavior as with a rigid body.
- Flexible body without RBB is considered.

- First step is to adjust the inertia properties of the body's FE-model according to the Adams rigid body. Body's mass and COG position are primarly adjusted by adding non structural mass (engine, driveline, tank, seats, doors, etc.). Moments of inertia are also slightly adjusted.

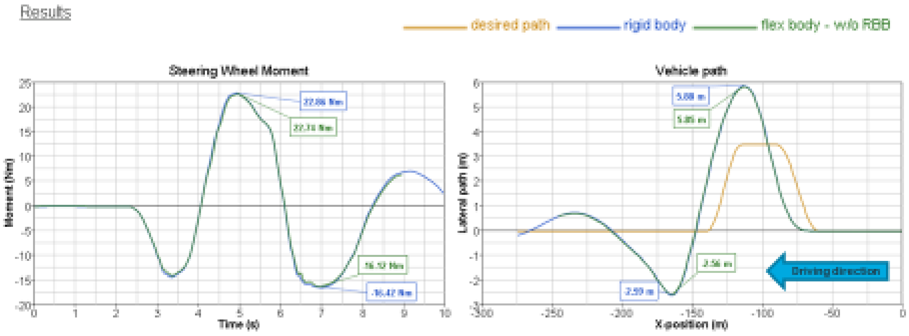
| | | | | rigid Body | | | flex Body w/o RBB | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Mass (kg) | | | | 1524,5 | | | 1524,4 | | | +0,1 kg | | |
| COG coordinates (mm) | X | | | 3244,4 | | | 3247,0 | | | -2,7 mm | | |
| | Y | | | 1,7 | | | -2,8 | | | +4,5 mm | | |
| | Z | | | 1784,4 | | | 1786,8 | | | -2,5 mm | | |
| Inertia tensor (to.mm^4) | Ixx | Ixy | Ixz | 9,4E+05 | 0 | 0 | 7,3E+05 | -1,4E+03 | -2,5E+05 | -21,0% | - | - |
| | Ixy | Iyy | Iyz | 0 | 4,2E+06 | 0 | -1,4E+03 | 3,9E+06 | -4,3E+03 | - | -7,8% | - |
| | Ixz | Iyz | Izz | 0 | 0 | 4,2E+06 | -2,5E+05 | -4,3E+03 | 4,0E+06 | - | - | -3,6% |

- Mass and COG of the flexible body fits the Adams rigid body
- Moments of inertia are lower, especially for Ixx

## 3. FIRST ANALYSIS: RIGID VS FLEXIBLE BODY (2/2)

Global Chassis Engineering

Results




- Vehicle's behavior is similar between rigid body and flexible body
- No solver parameter has been changed !!!



Go Further

## 4. INFLUENCE OF REAR BUMPER BEAM (1/8)

Global Chassis Engineering

Flexible Bodies comparison

- As the RBB is added to the body on the rear end of the vehicle, COG is located more on the rear of the vehicle. It leads to slightly higher YY and ZZ moments of inertia as well.

| | | | | flex Body w/o RBB | | | flex Body with RBB | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Mass (kg) | | | | 1524,4 | | | 1531,2 | | | +6,8 kg | | |
| COG coordinates (mm) | X | | | 3247,0 | | | 3260,2 | | | +13,2 mm | | |
| | Y | | | -2,8 | | | -2,8 | | | +0,0 mm | | |
| | Z | | | 1786,8 | | | 1784,6 | | | -2,2 mm | | |
| Inertia tensor (to.mm^4) | Ixx | Ixy | Ixz | 7,5E+05 | -1,4E+03 | -2,5E+05 | 7,5E+05 | -1,5E+03 | -2,4E+05 | +0,4% | +3,9% | -4,1% |
| | Ixy | Iyy | Iyz | -1,4E+03 | 3,9E+06 | -4,3E+03 | -1,5E+03 | 4,0E+06 | -4,3E+03 | +3,9% | +1,6% | -0,3% |
| | Ixz | Iyz | Izz | -2,5E+05 | -4,3E+03 | 4,0E+06 | -2,4E+05 | -4,3E+03 | 4,1E+06 | -4,1% | -0,3% | +1,5% |

- A static FE-analysis reveals that both bodies have the same roll stiffness



Go Further

## 4. INFLUENCE OF REAR BUMPER BEAM (2/8)

Global Chassis Engineering

Flexible Bodies comparison

- Flexible bodies are described by their normal modes. Modes frequencies of both bodies can be compared.
- 26 modes frequencies (over first 146 modes) deviate by more than 5 Hz.

Legend
Difference > +/- 5Hz

| mode | mode frequency (Hz) with RBB | mode frequency (Hz) without RBB | difference (Hz) | mode | mode frequency (Hz) with RBB | mode frequency (Hz) without RBB | difference (Hz) | mode | mode frequency (Hz) with RBB | mode frequency (Hz) without RBB | difference (Hz) | mode | mode frequency (Hz) with RBB | mode frequency (Hz) without RBB | difference (Hz) |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 7 | 8.44 | 8.44 | +0.0 | 42 | 72.78 | 72.79 | +0.0 | 77 | 326.02 | 332.78 | +6.8 | 112 | 1140.47 | 1180.50 | +39.8 |
| 8 | 8.73 | 8.73 | +0.0 | 43 | 75.51 | 75.59 | +0.1 | 78 | 346.53 | 348.88 | +2.3 | 113 | 1193.49 | 1197.58 | +4.1 |
| 9 | 14.95 | 14.95 | +0.0 | 44 | 81.48 | 81.62 | +0.1 | 79 | 363.18 | 367.98 | +4.8 | 114 | 1518.36 | 1517.33 | -1.0 |
| 10 | 15.52 | 15.52 | +0.0 | 45 | 86.60 | 87.24 | +0.6 | 80 | 384.20 | 384.72 | +0.5 | 115 | 1421.80 | 1422.00 | +0.2 |
| 11 | 17.48 | 17.48 | +0.0 | 46 | 90.75 | 90.77 | +0.0 | 81 | 400.80 | 397.92 | -2.8 | 116 | 1480.97 | 1480.98 | +0.0 |
| 12 | 17.48 | 17.48 | +0.0 | 47 | 93.00 | 93.17 | +0.1 | 82 | 409.08 | 434.78 | +25.7 | 117 | 1594.72 | 1595.01 | +0.3 |
| 13 | 17.92 | 17.92 | +0.0 | 48 | 96.90 | 97.07 | +0.1 | 83 | 430.60 | 443.99 | +13.8 | 118 | 1659.95 | 1661.65 | +1.7 |
| 14 | 18.43 | 18.47 | +0.0 | 49 | 99.75 | 99.78 | +0.0 | 84 | 437.00 | 445.88 | +8.8 | 119 | 1708.69 | 1715.51 | +6.8 |
| 15 | 18.94 | 18.95 | +0.0 | 50 | 112.55 | 112.65 | +0.1 | 85 | 446.08 | 454.75 | +8.7 | 120 | 1751.38 | 1755.37 | +4.0 |
| 16 | 19.42 | 19.45 | +0.0 | 51 | 117.75 | 117.85 | +0.1 | 86 | 458.79 | 462.75 | +4.0 | 121 | 2034.35 | 2023.24 | -11.1 |
| 17 | 20.48 | 20.49 | +0.1 | 52 | 125.27 | 125.36 | +0.1 | 87 | 469.12 | 480.92 | +11.8 | 122 | 2090.34 | 2167.98 | +77.7 |
| 18 | 22.05 | 22.06 | +0.0 | 53 | 127.15 | 127.55 | +0.4 | 88 | 484.37 | 487.78 | +3.4 | 123 | 3026.66 | 3028.45 | +1.8 |
| 19 | 22.60 | 22.60 | +0.0 | 54 | 128.12 | 128.41 | +0.3 | 89 | 499.87 | 499.82 | -0.0 | 124 | 3069.72 | 3070.39 | +0.7 |
| 20 | 24.88 | 24.97 | +0.1 | 55 | 135.09 | 135.48 | +0.4 | 90 | 515.80 | 516.33 | +0.5 | 125 | 3186.28 | 3186.82 | +0.5 |
| 21 | 25.64 | 25.65 | +0.0 | 56 | 136.70 | 137.11 | +0.4 | 91 | 533.79 | 541.42 | +7.6 | 126 | 3178.42 | 3178.56 | +0.1 |
| 22 | 26.80 | 26.81 | +0.0 | 57 | 145.40 | 147.12 | +1.7 | 92 | 547.71 | 552.23 | +4.5 | 127 | 3488.06 | 3487.22 | +1.1 |
| 23 | 27.66 | 27.66 | -0.0 | 58 | 146.25 | 147.49 | +1.2 | 93 | 556.22 | 560.85 | +4.6 | 128 | 3556.39 | 3558.91 | +2.5 |
| 24 | 28.62 | 28.65 | +0.0 | 59 | 153.77 | 153.85 | +0.1 | 94 | 575.23 | 576.12 | +0.9 | 129 | 3909.65 | 3909.92 | +0.3 |
| 25 | 29.55 | 29.59 | +0.0 | 60 | 165.35 | 165.54 | +0.2 | 95 | 603.48 | 602.18 | -1.3 | 130 | 4048.41 | 4048.93 | +0.5 |
| 26 | 30.25 | 30.26 | +0.0 | 61 | 171.15 | 171.18 | +0.0 | 96 | 621.44 | 623.48 | +2.0 | 131 | 5958.32 | 5958.40 | +1.1 |
| 27 | 30.44 | 30.44 | +0.0 | 62 | 178.85 | 178.85 | +0.0 | 97 | 687.53 | 687.85 | +0.3 | 132 | 6465.79 | 6465.87 | +0.1 |
| 28 | 30.87 | 30.87 | -0.0 | 63 | 184.10 | 184.89 | +0.8 | 98 | 727.42 | 728.53 | +1.1 | 133 | 6672.42 | 6673.41 | +0.0 |
| 29 | 31.83 | 31.83 | -0.0 | 64 | 204.10 | 205.11 | +1.0 | 99 | 768.02 | 788.48 | +20.6 | 134 | 6820.69 | 6850.55 | +6.9 |
| 30 | 34.11 | 34.11 | +0.0 | 65 | 220.00 | 220.66 | +0.7 | 100 | 788.18 | 798.47 | +10.3 | 135 | 7801.39 | 7801.93 | +0.5 |
| 31 | 38.88 | 38.98 | +0.1 | 66 | 235.01 | 235.82 | +0.8 | 101 | 803.79 | 811.24 | +7.4 | 136 | 7982.34 | 7983.81 | +1.4 |
| 32 | 40.71 | 40.77 | +0.1 | 67 | 248.70 | 251.19 | +2.5 | 102 | 811.54 | 817.90 | +6.4 | 137 | 12486.75 | 12486.75 | +0.0 |
| 33 | 46.64 | 46.91 | +0.3 | 68 | 253.44 | 254.58 | +1.1 | 103 | 828.17 | 831.62 | +3.5 | 138 | 12493.07 | 12493.07 | +0.0 |
| 34 | 47.29 | 47.30 | +0.0 | 69 | 261.55 | 268.35 | +6.8 | 104 | 836.05 | 836.75 | +0.7 | 139 | 22275.23 | 22275.24 | +0.0 |
| 35 | 52.75 | 52.85 | +0.1 | 70 | 267.45 | 275.58 | +8.1 | 105 | 873.81 | 889.46 | +15.7 | 140 | 22522.68 | 22522.69 | +0.0 |
| 36 | 55.80 | 55.82 | +0.0 | 71 | 276.02 | 277.07 | +1.0 | 106 | 888.85 | 902.90 | +8.2 | 141 | 37908.26 | 37908.27 | +0.1 |
| 37 | 59.46 | 58.72 | -0.7 | 72 | 297.09 | 300.00 | +0.8 | 107 | 920.40 | 932.09 | +11.7 | 142 | 37980.76 | 37980.76 | +0.0 |
| 38 | 61.83 | 61.88 | +0.0 | 73 | 303.07 | 304.10 | +0.8 | 108 | 991.75 | 988.37 | -3.4 | 143 | 42861.83 | 42861.83 | +0.1 |
| 39 | 65.73 | 65.59 | -0.1 | 74 | 301.70 | 320.45 | +18.7 | 109 | 1027.49 | 1072.50 | +45.1 | 144 | 42873.89 | 42873.89 | +0.0 |
| 40 | 65.97 | 66.14 | +0.2 | 75 | 313.05 | 325.69 | +12.7 | 110 | 1063.47 | 1079.02 | +15.5 | 145 | 46584.19 | 46584.24 | +0.1 |
| 41 | 71.84 | 72.05 | +0.2 | 76 | 323.58 | 328.12 | +4.5 | 111 | 1077.38 | 1104.83 | +27.4 | 146 | 49400.39 | 49400.40 | +0.0 |



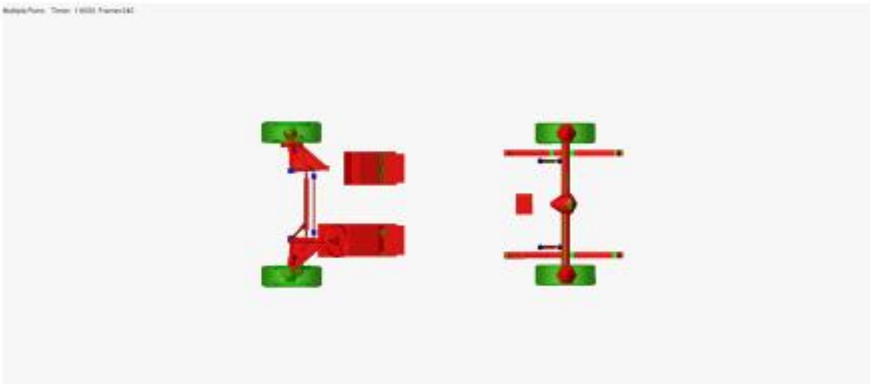
## 4. INFLUENCE OF REAR BUMPER BEAM (3/8)

Global Chassis Engineering

Results – Animation

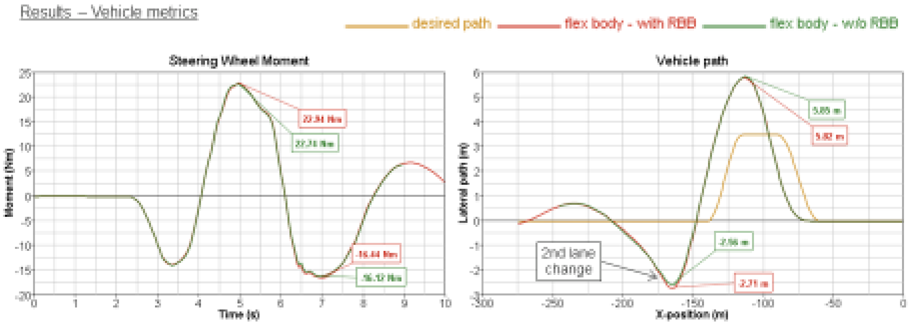
(top view, flexible body is not shown for a better comparison)

flex body - with RBB  flex body - w/o RBB



## 4. INFLUENCE OF REAR BUMPER BEAM (4/8)

Results – Vehicle metrics

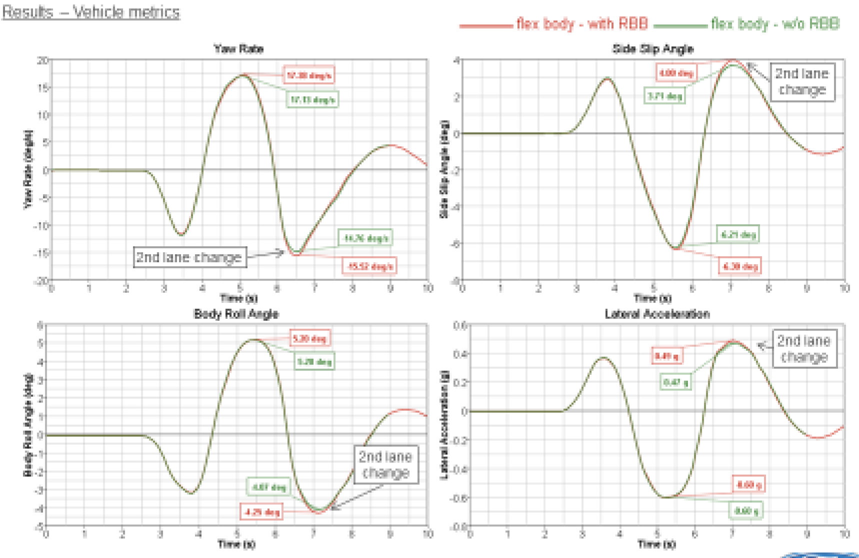



- With the RBB, the driver needs to apply a higher moment at the steering wheel torque (+0.2 / +0.3 Nm)
- Model with RBB leads to more Y-deviation in the second lane change by 15 cm



## 4. INFLUENCE OF REAR BUMPER BEAM (5/8)

Results – Vehicle metrics

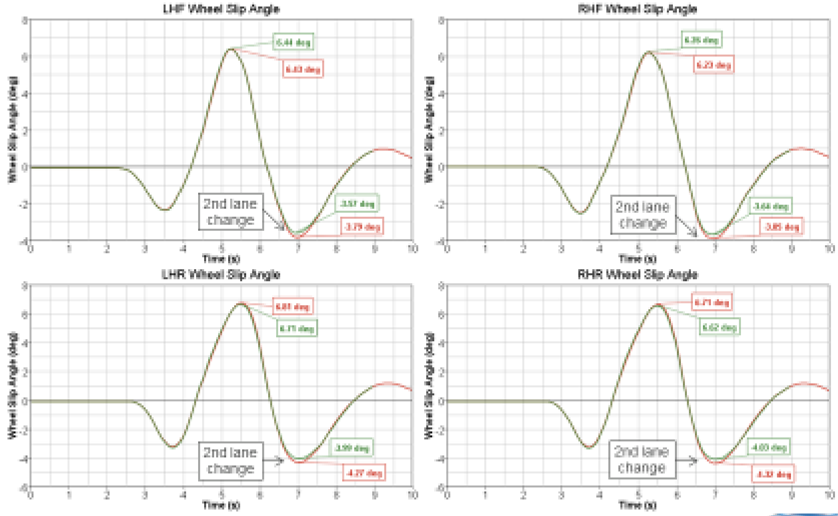


## 4. INFLUENCE OF REAR BUMPER BEAM (6/8)

Global Chassis Engineering

Results – Wheel slip angle

flex body - with RBB — flex body - w/o RBB



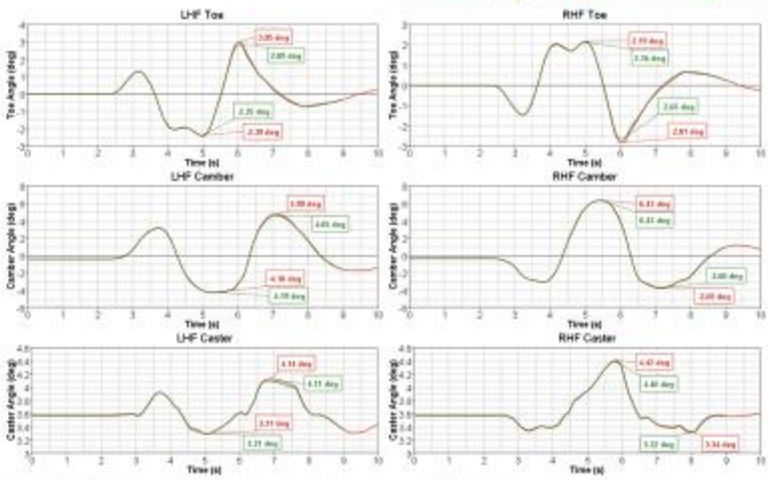



## 4. INFLUENCE OF REAR BUMPER BEAM (7/8)

Global Chassis Engineering

Results – Front Axle toe, camber & caster

flex body - with RBB — flex body - w/o RBB

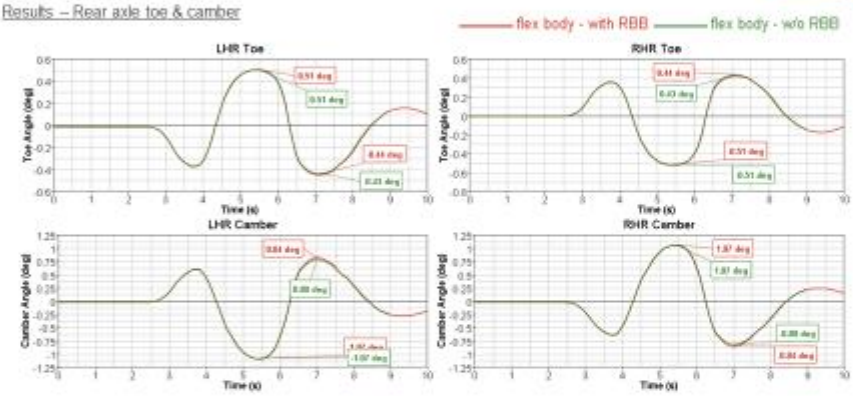



- Front toe is slightly higher with RBB during second lane change (+0.15 deg)
- Camber is also slightly higher, especially on LHS (+0.25 deg)

## 4. INFLUENCE OF REAR BUMPER BEAM (8/8)

Global Chassis Engineering

Results – Rear axle toe & camber




- Rear toe is the same between both vehicles
- Rear camber is a bit bigger with RBB during second lane change (+0.04 deg)



## 5. CONCLUSION & NEXT STEPS

Global Chassis Engineering

**Conclusion**

- Flexible body works together with Pacejka tire model as long as the inertia properties of the flexible body fits the rigid body that it replaces
- Adams simulation is very sensitive to any change in the flexible body and in solver parameters
- Vehicle's behavior looks better without RBB. Difference is however minimal and can be seen during the second lane change

**Possible next step**

- Run VDA Frequency Response analysis
  - *"provide steering input that spans the entire frequency range of a human driver, roughly 0 to 3 Hz"* (source: Adams Help)
  - would better reveal the dynamic behavior of the vehicle and the influence of the RBB
  - analysis not working yet

## 6. ADDITIONAL ANALYSIS: BODY STRENGTHENING (1/14)

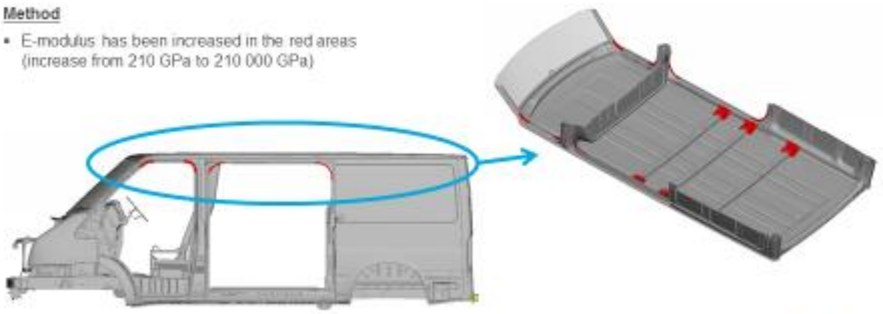
Global Chassis Engineering

**Idea**

- Strengthen the body around the weakest areas to see the influence on the vehicle during double change
- Following areas have been identified by Durability team as the weakest ones:
  - Top of the A pillars
  - Top of the B pillars
  - Top of the C pillars
  - All of the roof rails (which are just assembled with foam)

**Method**

- E-modulus has been increased in the red areas (increase from 210 GPa to 210 000 GPa)





## 6. ADDITIONAL ANALYSIS: BODY STRENGTHENING (2/14)

Global Chassis Engineering

Flexible Bodies comparison: **body w/o RBB vs stiffened body w/o RBB**

Legend
Difference > +/- 5Hz

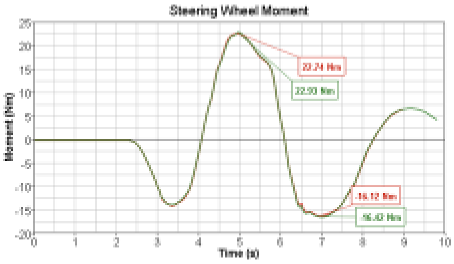
| mode | mode frequency (Hz) | | difference (Hz) | mode | mode frequency (Hz) | | difference (Hz) | mode | mode frequency (Hz) | | difference (Hz) | mode | mode frequency (Hz) | | difference (Hz) |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | w/o RBB | stiffened w/o RBB | | | w/o RBB | stiffened w/o RBB | | | w/o RBB | stiffened w/o RBB | | | w/o RBB | stiffened w/o RBB | |
| 7 | 8,44 | 8,44 | +0,0 | 42 | 72,79 | 72,90 | +0,1 | 77 | 326,02 | 340,00 | +14,0 | 112 | 1140,47 | 1190,00 | +49,5 |
| 8 | 8,78 | 8,78 | -0,0 | 43 | 75,59 | 75,60 | +0,0 | 78 | 346,56 | 350,00 | +3,5 | 113 | 1196,48 | 1230,00 | +33,5 |
| 9 | 14,86 | 14,86 | -0,0 | 44 | 81,62 | 81,40 | -0,2 | 79 | 365,18 | 378,00 | +12,8 | 114 | 1216,56 | 1240,00 | +23,4 |
| 10 | 15,52 | 15,50 | -0,0 | 45 | 87,24 | 86,20 | -1,0 | 80 | 384,30 | 387,00 | +2,8 | 115 | 1421,80 | 1430,00 | +8,2 |
| 11 | 17,45 | 17,40 | -0,0 | 46 | 90,77 | 90,50 | -0,3 | 81 | 400,30 | 400,00 | -0,3 | 116 | 1465,57 | 1470,00 | +4,4 |
| 12 | 17,46 | 17,50 | +0,0 | 47 | 95,17 | 95,20 | +0,0 | 82 | 405,05 | 439,00 | +34,0 | 117 | 1594,72 | 1620,00 | +25,3 |
| 13 | 17,92 | 17,90 | -0,0 | 48 | 97,07 | 98,30 | +1,2 | 83 | 430,66 | 445,00 | +14,4 | 118 | 1659,86 | 1670,00 | +10,1 |
| 14 | 18,47 | 19,10 | +0,6 | 49 | 99,78 | 100,00 | +0,2 | 84 | 437,00 | 455,00 | +18,0 | 119 | 1758,66 | 1770,00 | +11,3 |
| 15 | 18,86 | 19,20 | +0,3 | 50 | 112,68 | 112,00 | -0,6 | 85 | 446,06 | 459,00 | +12,9 | 120 | 1771,38 | 1780,00 | +8,6 |
| 16 | 19,45 | 20,10 | +0,7 | 51 | 117,85 | 118,00 | +0,1 | 86 | 458,79 | 470,00 | +11,2 | 121 | 2034,35 | 2030,00 | -4,4 |
| 17 | 20,49 | 21,40 | +0,9 | 52 | 125,36 | 126,00 | +0,6 | 87 | 469,12 | 485,00 | +15,9 | 122 | 2090,24 | 2170,00 | +79,8 |
| 18 | 21,06 | 21,50 | +0,4 | 53 | 127,55 | 128,00 | +0,5 | 88 | 482,37 | 496,00 | +13,6 | 123 | 3026,66 | 3030,00 | +3,3 |
| 19 | 21,66 | 22,00 | +0,4 | 54 | 128,61 | 129,00 | +0,4 | 89 | 499,57 | 509,00 | +9,4 | 124 | 3069,71 | 3070,00 | +0,3 |
| 20 | 24,97 | 25,40 | +0,4 | 55 | 135,88 | 135,00 | -0,9 | 90 | 525,80 | 540,00 | +14,2 | 125 | 3166,16 | 3170,00 | +3,8 |
| 21 | 25,65 | 26,30 | +0,7 | 56 | 137,11 | 138,00 | +0,9 | 91 | 533,79 | 547,00 | +13,2 | 126 | 3173,45 | 3240,00 | +66,6 |
| 22 | 26,51 | 26,60 | +0,1 | 57 | 147,12 | 148,00 | +0,9 | 92 | 547,71 | 555,00 | +7,3 | 127 | 3436,04 | 3440,00 | +4,0 |
| 23 | 27,66 | 27,90 | +0,2 | 58 | 147,49 | 148,00 | +0,5 | 93 | 556,12 | 575,00 | +18,9 | 128 | 3536,39 | 3540,00 | +3,6 |
| 24 | 28,66 | 28,80 | +0,2 | 59 | 150,85 | 151,00 | +0,1 | 94 | 575,26 | 588,00 | +12,8 | 129 | 3909,60 | 3970,00 | +60,4 |
| 25 | 29,58 | 29,10 | -0,5 | 60 | 160,54 | 160,00 | -0,5 | 95 | 600,48 | 607,00 | +6,5 | 130 | 4049,41 | 4240,00 | +190,6 |
| 26 | 30,26 | 30,40 | +0,1 | 61 | 171,16 | 173,00 | +1,8 | 96 | 631,44 | 660,00 | +28,6 | 131 | 5956,32 | 5960,00 | +3,7 |
| 27 | 30,44 | 30,50 | +0,1 | 62 | 178,65 | 181,00 | +2,3 | 97 | 697,55 | 701,00 | +3,5 | 132 | 6465,79 | 6470,00 | +4,2 |
| 28 | 30,67 | 30,70 | +0,0 | 63 | 187,89 | 186,00 | -1,9 | 98 | 727,42 | 730,00 | +2,6 | 133 | 6472,41 | 6470,00 | -2,4 |
| 29 | 31,84 | 32,10 | +0,3 | 64 | 206,11 | 207,00 | +0,9 | 99 | 760,00 | 789,00 | +28,0 | 134 | 6841,66 | 6850,00 | +8,3 |
| 30 | 34,11 | 34,10 | -0,0 | 65 | 226,66 | 227,00 | +0,3 | 100 | 788,18 | 799,00 | +10,8 | 135 | 7981,18 | 7980,00 | -1,2 |
| 31 | 38,56 | 40,30 | +1,7 | 66 | 235,52 | 235,00 | -0,5 | 101 | 806,79 | 812,00 | +5,2 | 136 | 7982,34 | 7980,00 | -2,3 |
| 32 | 40,77 | 45,00 | +4,2 | 67 | 251,19 | 252,00 | +0,8 | 102 | 811,84 | 821,00 | +9,2 | 137 | 12466,75 | 12500,00 | +33,2 |
| 33 | 46,91 | 45,70 | -1,2 | 68 | 254,56 | 259,00 | +4,4 | 103 | 828,17 | 834,00 | +5,8 | 138 | 12495,07 | 12500,00 | +4,9 |
| 34 | 47,30 | 48,90 | +1,6 | 69 | 268,35 | 268,00 | -0,4 | 104 | 836,05 | 844,00 | +7,9 | 139 | 22275,15 | 22300,00 | +24,8 |
| 35 | 52,85 | 51,60 | -1,3 | 70 | 275,58 | 277,00 | +1,4 | 105 | 871,81 | 890,00 | +18,2 | 140 | 22322,68 | 22300,00 | -22,7 |
| 36 | 55,52 | 57,30 | +1,8 | 71 | 277,37 | 278,00 | +0,6 | 106 | 896,35 | 931,00 | +34,6 | 141 | 37906,16 | 37900,00 | -6,2 |
| 37 | 58,72 | 58,70 | -0,0 | 72 | 300,38 | 300,00 | -0,4 | 107 | 930,48 | 958,00 | +27,6 | 142 | 37992,75 | 38000,00 | +28,2 |
| 38 | 61,86 | 63,10 | +1,2 | 73 | 304,24 | 307,00 | +2,8 | 108 | 991,75 | 992,00 | +0,2 | 143 | 42841,55 | 42800,00 | -41,5 |
| 39 | 65,59 | 66,00 | +0,4 | 74 | 320,45 | 323,00 | +2,5 | 109 | 1027,49 | 1080,00 | +52,5 | 144 | 42675,95 | 42700,00 | +24,1 |
| 40 | 66,14 | 67,80 | +1,7 | 75 | 325,66 | 326,00 | +0,3 | 110 | 1066,47 | 1100,00 | +33,5 | 145 | 39964,16 | 39900,00 | -63,8 |
| 41 | 72,05 | 72,50 | +0,5 | 76 | 328,12 | 332,00 | +3,9 | 111 | 1077,38 | 1120,00 | +42,6 | 146 | 49400,39 | 49400,00 | -0,4 |

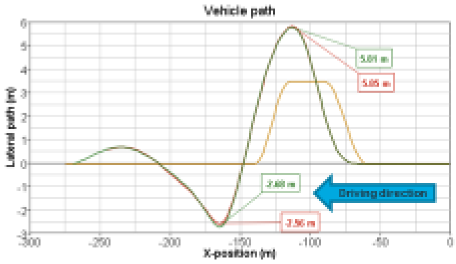
## 6. ADDITIONAL ANALYSIS: BODY STRENGTHENING (3/14)

Global Chassis Engineering

Results: **body w/o RBB vs stiffened body w/o RBB**

flex body - w/o RBB — flex body stiffened - w/o RBB — desired path




- With a stiffened body, the driver needs to apply a higher moment at the steering wheel torque (+0.2 / +0.3 Nm)
- Model with stiffened body leads to more Y-deviation in the second lane change by 10 cm

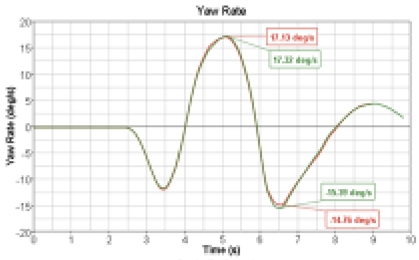


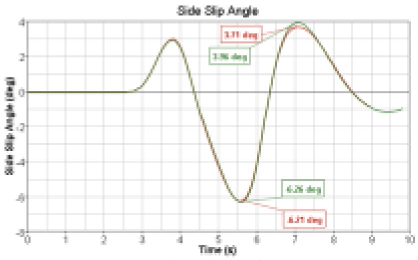
## 6. ADDITIONAL ANALYSIS: BODY STRENGTHENING (4/14)

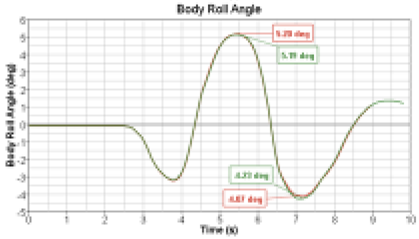
Global Chassis Engineering

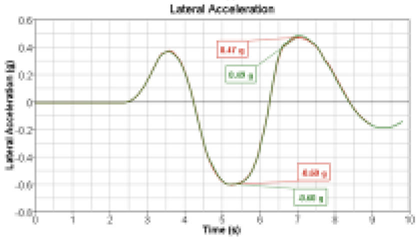
Results: **body w/o RBB vs stiffened body w/o RBB**

flex body - w/o RBB — flex body stiffened - w/o RBB



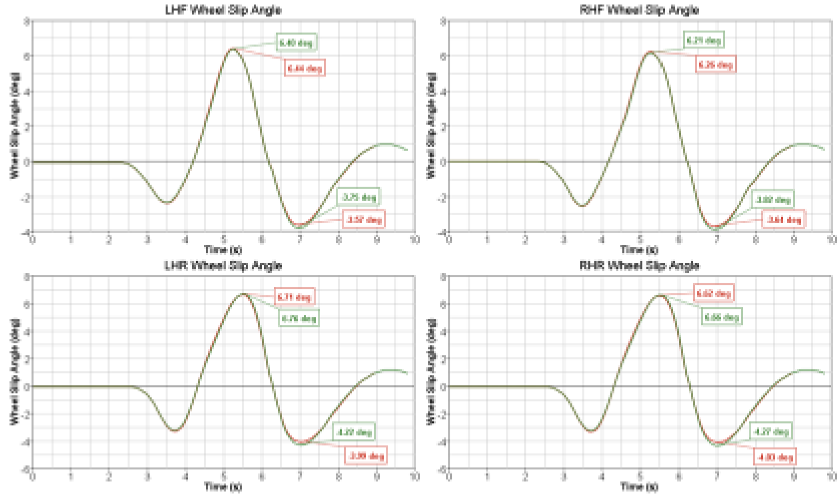

- Yaw rate, side slip angle, body roll angle increase with the stiffened body
- Stiffened body almost doesn't affect lateral acceleration

## 6. ADDITIONAL ANALYSIS: BODY STRENGTHENING (5/14)

Global Chassis Engineering

Results: **body w/o RBB vs stiffened body w/o RBB**

flex body - w/o RBB — flex body stiffened - w/o RBB




- A stiffened body leads to higher wheel slip angle, especially during the second lane change

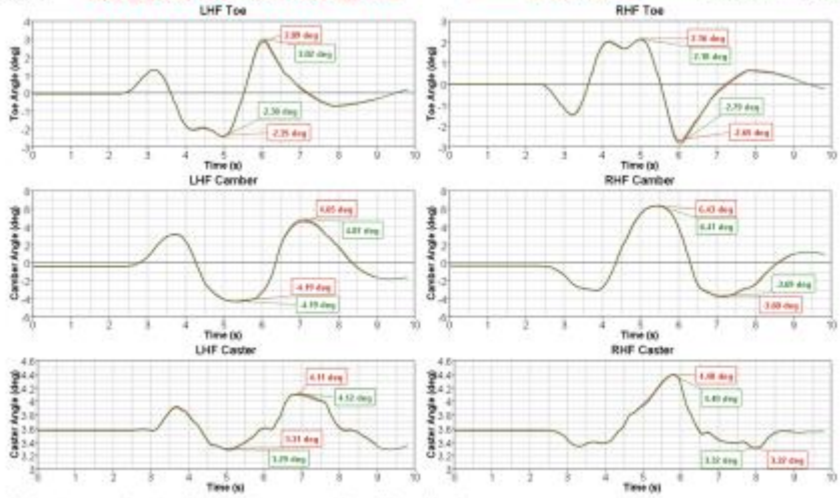


## 6. ADDITIONAL ANALYSIS: BODY STRENGTHENING (6/14)

Global Chassis Engineering

Results: **body w/o RBB vs stiffened body w/o RBB**

flex body - w/o RBB — flex body stiffened - w/o RBB



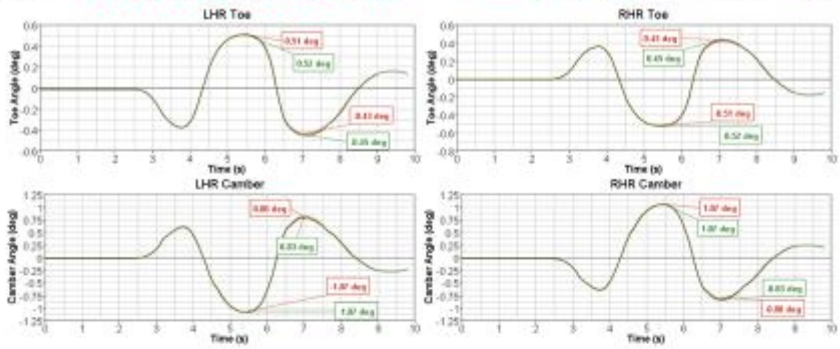

- On front suspension, toe & camber increase with a stiffened body
- Caster is not affected

## 6. ADDITIONAL ANALYSIS: BODY STRENGTHENING (7/14)

Global Chassis Engineering

Results: **body w/o RBB vs stiffened body w/o RBB**

flex body - w/o RBB — flex body stiffened - w/o RBB




- A stiffened body doesn't affect toe and camber on rear suspension



## 6. ADDITIONAL ANALYSIS: BODY STRENGTHENING (8/14)

Global Chassis Engineering

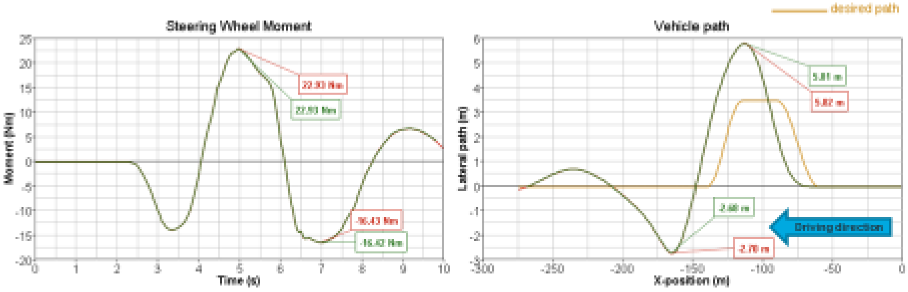
Flexible Bodies comparison: **body with RBB vs stiffened body with RBB**

Legend
Difference > +/- 5Hz

| mode | mode frequency (Hz) | | difference | mode | mode frequency (Hz) | | difference | mode | mode frequency (Hz) | | difference | mode | mode frequency (Hz) | | difference |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | with RBB | stiffened with RBB | (Hz) | | with RBB | stiffened with RBB | (Hz) | | with RBB | stiffened with RBB | (Hz) | | with RBB | stiffened with RBB | (Hz) |
| 7 | 8,44 | 8,44 | +0,0 | 42 | 72,76 | 72,85 | +0,1 | 77 | 326,02 | 340,39 | +14,4 | 112 | 1140,47 | 1194,09 | +53,6 |
| 8 | 8,73 | 8,73 | +0,0 | 43 | 75,51 | 75,58 | +0,1 | 78 | 346,59 | 350,31 | +3,7 | 113 | 1193,49 | 1221,37 | +27,9 |
| 9 | 14,95 | 14,95 | +0,0 | 44 | 81,48 | 81,25 | -0,2 | 79 | 363,15 | 375,08 | +11,9 | 114 | 1205,26 | 1238,79 | +33,4 |
| 10 | 15,53 | 15,53 | +0,0 | 45 | 95,60 | 95,18 | -0,4 | 80 | 384,20 | 387,01 | +2,8 | 115 | 1421,80 | 1426,37 | +4,5 |
| 11 | 17,43 | 17,44 | +0,0 | 46 | 90,75 | 90,47 | -0,3 | 81 | 400,20 | 399,62 | -0,5 | 116 | 1462,57 | 1466,92 | +4,4 |
| 12 | 17,46 | 17,47 | +0,0 | 47 | 88,02 | 88,18 | +0,2 | 82 | 405,08 | 489,36 | +84,3 | 117 | 1584,72 | 1623,33 | +38,6 |
| 13 | 17,52 | 17,53 | +0,0 | 48 | 86,83 | 88,00 | +1,4 | 83 | 430,68 | 445,90 | +14,8 | 118 | 1659,93 | 1669,32 | +9,4 |
| 14 | 18,42 | 19,08 | +0,6 | 49 | 99,75 | 100,15 | +0,4 | 84 | 437,00 | 454,68 | +17,7 | 119 | 1738,69 | 1747,48 | +8,8 |
| 15 | 18,84 | 19,23 | +0,4 | 50 | 112,83 | 112,68 | -0,2 | 85 | 444,06 | 456,73 | +12,7 | 120 | 1751,18 | 1761,48 | +10,3 |
| 16 | 19,42 | 20,08 | +0,6 | 51 | 117,78 | 118,38 | +0,6 | 86 | 458,79 | 469,99 | +11,2 | 121 | 2014,85 | 2026,89 | +7,8 |
| 17 | 20,48 | 21,48 | +1,0 | 52 | 120,27 | 120,58 | +0,3 | 87 | 469,13 | 483,33 | +14,2 | 122 | 2090,12 | 2168,17 | +77,8 |
| 18 | 22,03 | 22,46 | +0,4 | 53 | 127,12 | 128,16 | +1,0 | 88 | 484,27 | 497,96 | +13,6 | 123 | 2025,90 | 2033,00 | +7,1 |
| 19 | 22,63 | 23,09 | +0,4 | 54 | 128,42 | 129,03 | +0,6 | 89 | 498,57 | 509,27 | +10,7 | 124 | 2069,71 | 2074,96 | +5,2 |
| 20 | 24,83 | 25,39 | +0,6 | 55 | 135,49 | 135,47 | -0,0 | 90 | 525,80 | 540,25 | +14,5 | 125 | 2165,16 | 2168,44 | +3,3 |
| 21 | 25,64 | 26,29 | +0,6 | 56 | 136,70 | 137,62 | +0,9 | 91 | 533,79 | 547,29 | +13,5 | 126 | 2171,43 | 2242,73 | +69,3 |
| 22 | 26,50 | 26,58 | +0,1 | 57 | 145,40 | 147,58 | +2,2 | 92 | 547,71 | 554,94 | +7,2 | 127 | 2436,04 | 2437,24 | +1,2 |
| 23 | 27,68 | 27,82 | +0,1 | 58 | 146,24 | 147,68 | +1,4 | 93 | 556,22 | 575,48 | +19,2 | 128 | 2836,89 | 2839,08 | +2,6 |
| 24 | 28,62 | 28,78 | +0,2 | 59 | 153,77 | 154,22 | +0,4 | 94 | 575,28 | 588,48 | +13,2 | 129 | 2809,63 | 2871,48 | +61,8 |
| 25 | 28,50 | 29,12 | -0,4 | 60 | 163,85 | 164,20 | +0,3 | 95 | 603,48 | 607,32 | +3,8 | 130 | 4049,62 | 4235,90 | +186,3 |
| 26 | 30,28 | 30,41 | +0,2 | 61 | 171,18 | 172,78 | +1,6 | 96 | 621,48 | 659,73 | +38,3 | 131 | 5956,62 | 5958,58 | +2,1 |
| 27 | 30,44 | 30,58 | +0,1 | 62 | 178,81 | 180,04 | +1,2 | 97 | 697,58 | 701,28 | +3,6 | 132 | 6083,79 | 6088,21 | +4,4 |
| 28 | 30,67 | 30,68 | +0,0 | 63 | 184,13 | 185,93 | +1,8 | 98 | 727,62 | 730,33 | +2,7 | 133 | 6473,02 | 6473,28 | +0,1 |
| 29 | 31,85 | 32,12 | +0,2 | 64 | 204,12 | 206,58 | +2,5 | 99 | 763,02 | 789,36 | +26,3 | 134 | 6942,69 | 6950,77 | +8,1 |
| 30 | 34,11 | 34,12 | +0,0 | 65 | 223,00 | 225,76 | +2,8 | 100 | 788,15 | 799,61 | +11,5 | 135 | 7931,18 | 7931,65 | +0,5 |
| 31 | 38,38 | 40,34 | +2,0 | 66 | 235,01 | 235,33 | +0,3 | 101 | 803,79 | 811,93 | +8,1 | 136 | 7982,54 | 7984,04 | +1,5 |
| 32 | 40,73 | 44,96 | +4,2 | 67 | 248,70 | 252,25 | +3,5 | 102 | 811,84 | 820,82 | +9,0 | 137 | 12466,75 | 12467,41 | +0,7 |
| 33 | 46,64 | 45,71 | -0,9 | 68 | 253,44 | 258,87 | +5,4 | 103 | 828,17 | 834,34 | +6,2 | 138 | 12483,07 | 12484,22 | +1,2 |
| 34 | 47,28 | 48,87 | +1,6 | 69 | 261,58 | 268,17 | +6,6 | 104 | 836,05 | 844,93 | +8,9 | 139 | 22275,26 | 22277,60 | +2,3 |
| 35 | 52,79 | 51,58 | -1,2 | 70 | 267,43 | 277,24 | +9,8 | 105 | 873,81 | 889,48 | +15,7 | 140 | 22321,68 | 22326,10 | +4,4 |
| 36 | 55,30 | 57,18 | +1,9 | 71 | 276,82 | 278,17 | +1,4 | 106 | 898,85 | 992,02 | +93,7 | 141 | 37904,16 | 37910,75 | +6,6 |
| 37 | 59,40 | 58,72 | -0,7 | 72 | 287,09 | 289,72 | +2,6 | 107 | 920,18 | 988,36 | +67,5 | 142 | 37990,79 | 37992,38 | +1,6 |
| 38 | 61,83 | 63,10 | +1,3 | 73 | 300,47 | 307,08 | +6,6 | 108 | 991,75 | 992,43 | +0,7 | 143 | 42841,35 | 42842,88 | +1,5 |
| 39 | 65,78 | 65,99 | +0,3 | 74 | 301,74 | 322,88 | +21,1 | 109 | 1027,09 | 1075,02 | +47,9 | 144 | 42875,86 | 42877,59 | +1,7 |
| 40 | 65,87 | 67,81 | +1,9 | 75 | 313,02 | 327,95 | +14,9 | 110 | 1063,47 | 1104,60 | +41,1 | 145 | 46964,13 | 46966,05 | +1,9 |
| 41 | 71,84 | 72,58 | +0,7 | 76 | 323,58 | 331,88 | +8,3 | 111 | 1077,26 | 1120,08 | +42,8 | 146 | 49400,26 | 49400,58 | +0,3 |

## 6. ADDITIONAL ANALYSIS: BODY STRENGTHENING (9/14)

Results: **body with RBB vs stiffened body with RBB**

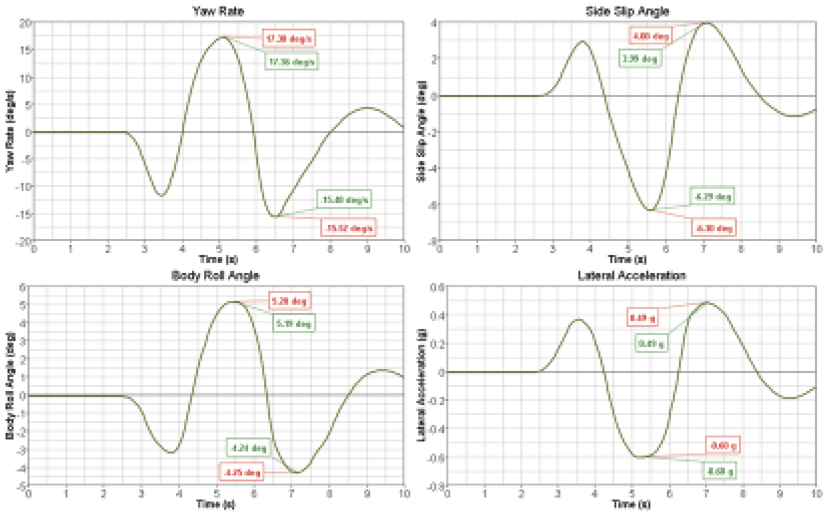



- A stiffened body doesn't affect the moment at the steering wheel
- Vehicle path is a bit affected, but difference is insignificant



## 6. ADDITIONAL ANALYSIS: BODY STRENGTHENING (10/14)

Results: **body with RBB vs stiffened body with RBB**

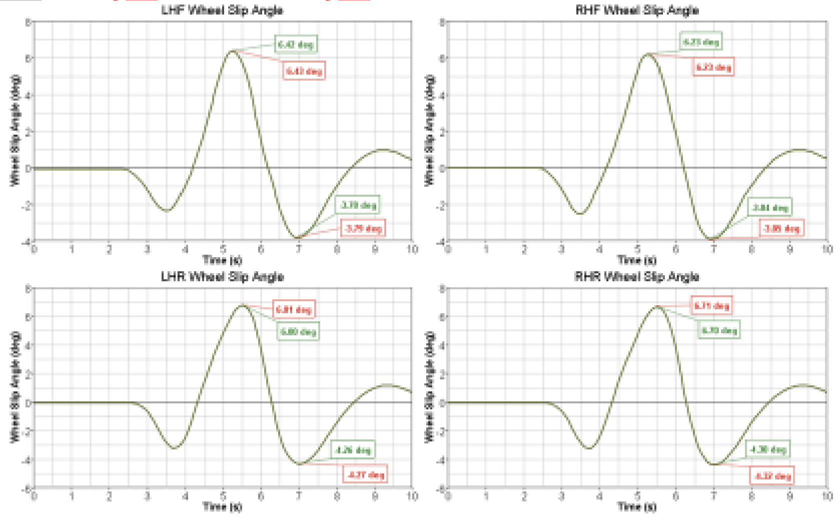



- Influence of a stiffened body is insignifcant

## 6. ADDITIONAL ANALYSIS: BODY STRENGTHENING (11/14)

Global Chassis Engineering

Results: **body with RBB vs stiffened body with RBB** —— flex body - with RBB —— flex body stiffened - with RBB



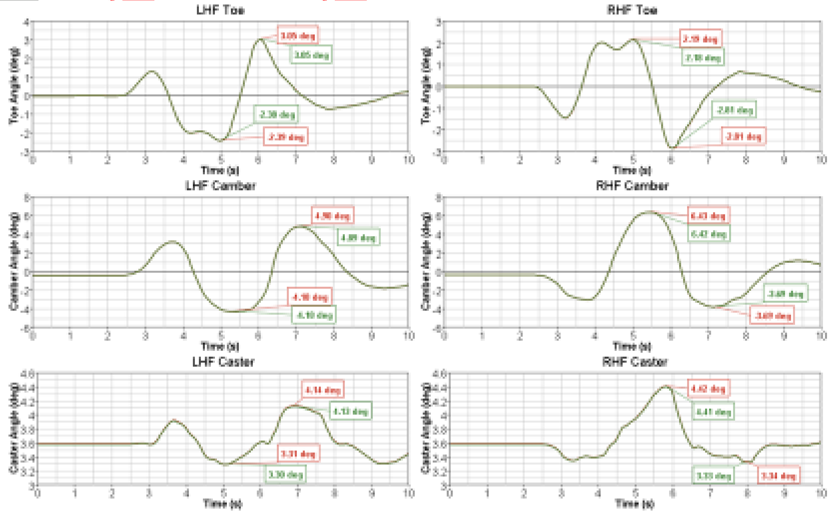

- Influence of a stiffened body is insignifcant



## 6. ADDITIONAL ANALYSIS: BODY STRENGTHENING (12/14)

Global Chassis Engineering

Results: **body with RBB vs stiffened body with RBB** —— flex body - with RBB —— flex body stiffened - with RBB

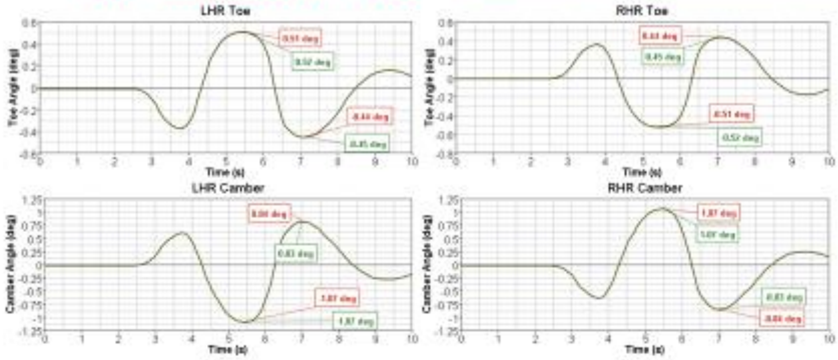



- Influence of a stiffened body is insignifcant

## 6. ADDITIONAL ANALYSIS: BODY STRENGTHENING (13/14)

Global Chassis Engineering

Results: **body with RBB vs stiffened body with RBB**




- Influence of a stiffened body is insignifcant



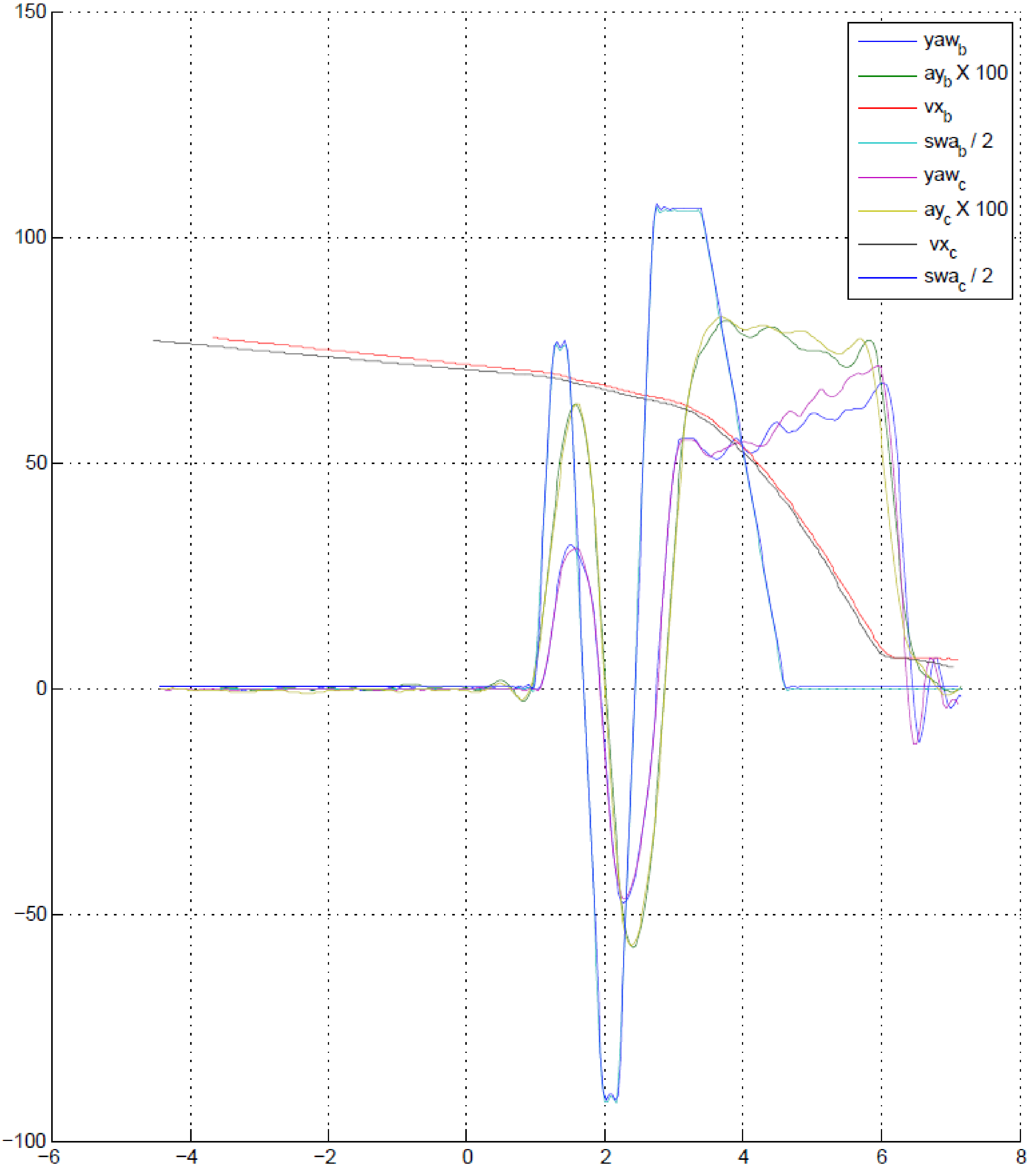
## 6. ADDITIONAL ANALYSIS: BODY STRENGTHENING (14/14)

Global Chassis Engineering

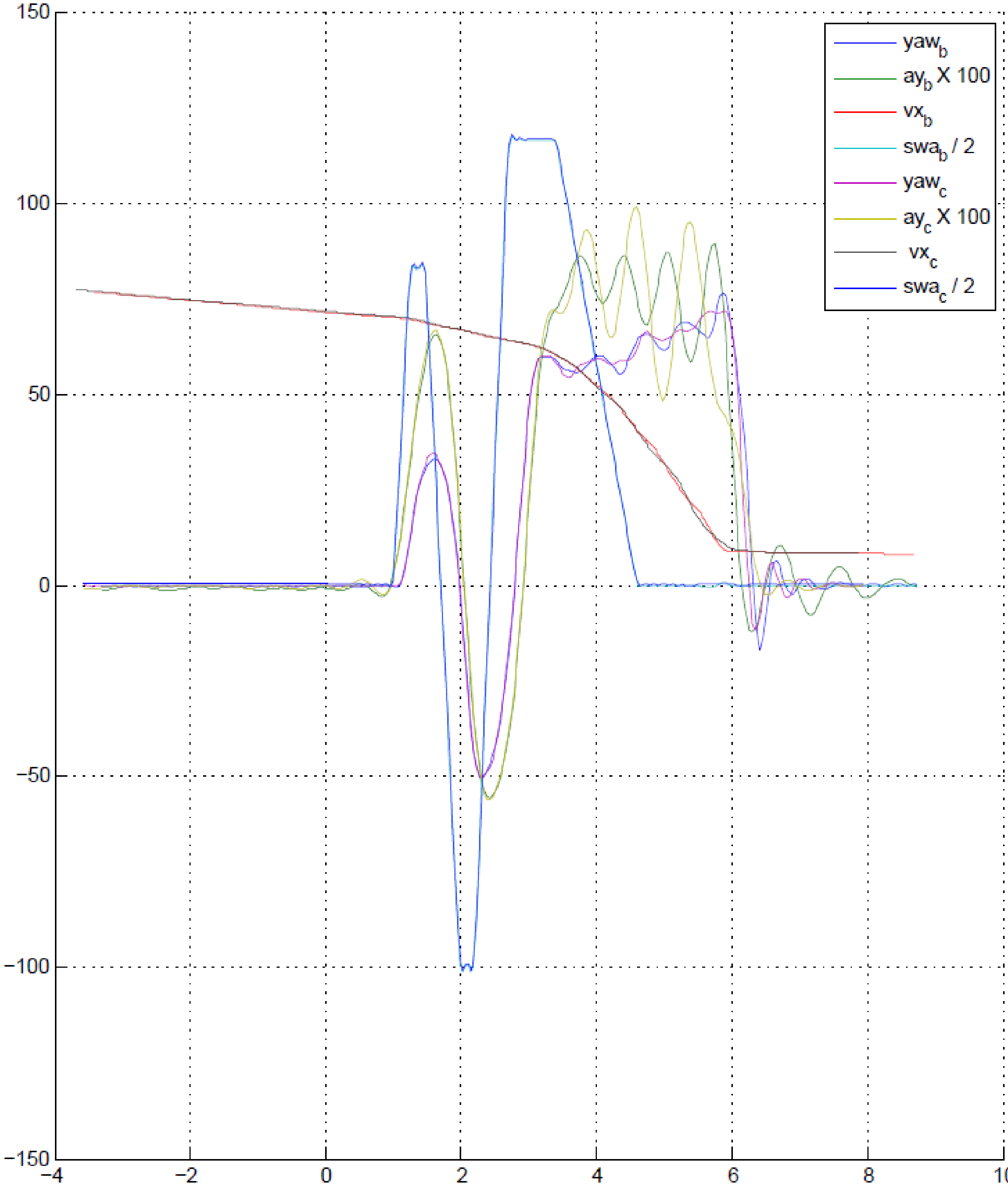
Conclusion

- When there isn't any RBB, the influence of stiffening the body at the top of A, B & C pillars and at the roof rails can be seen during double lane change event. As the body is less flexible, the vehicle has a worse behavior.
- By adding the RBB, it can be seen that stiffening the body doesn't affect the vehicle's behavior, which leads to the conclusion that adding a RBB dramatically increases the rigidity of the body.
- Once again, double lane change maneuver might not be dynamic enough and a more dynamic analysis like VDA Frequency Response might show the effect of the RBB even better.

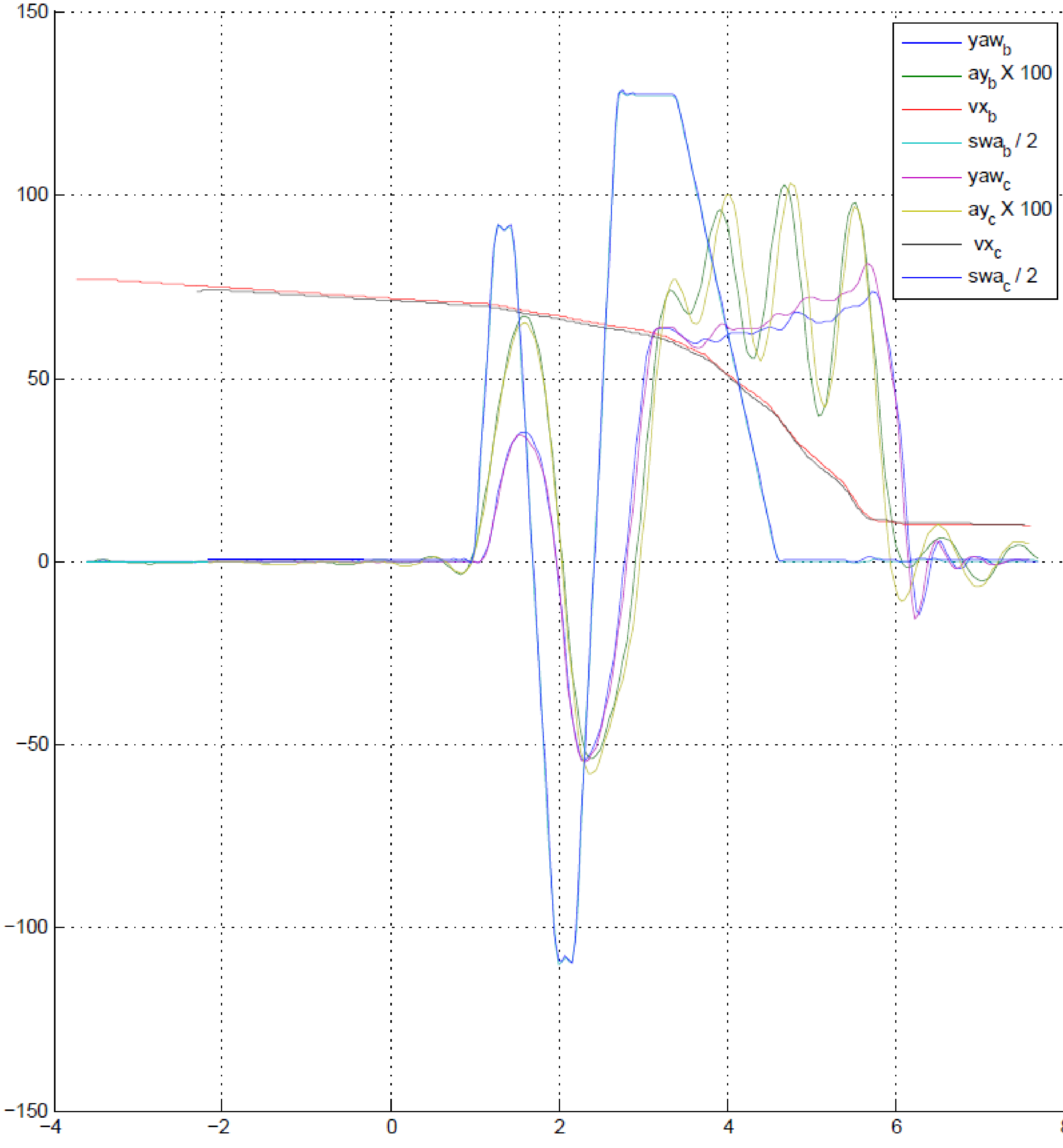
**Yaw / Lateral Acceleration vs Steering Wheel Angle with Low Steering Amplitude Prior to Body Modifications**



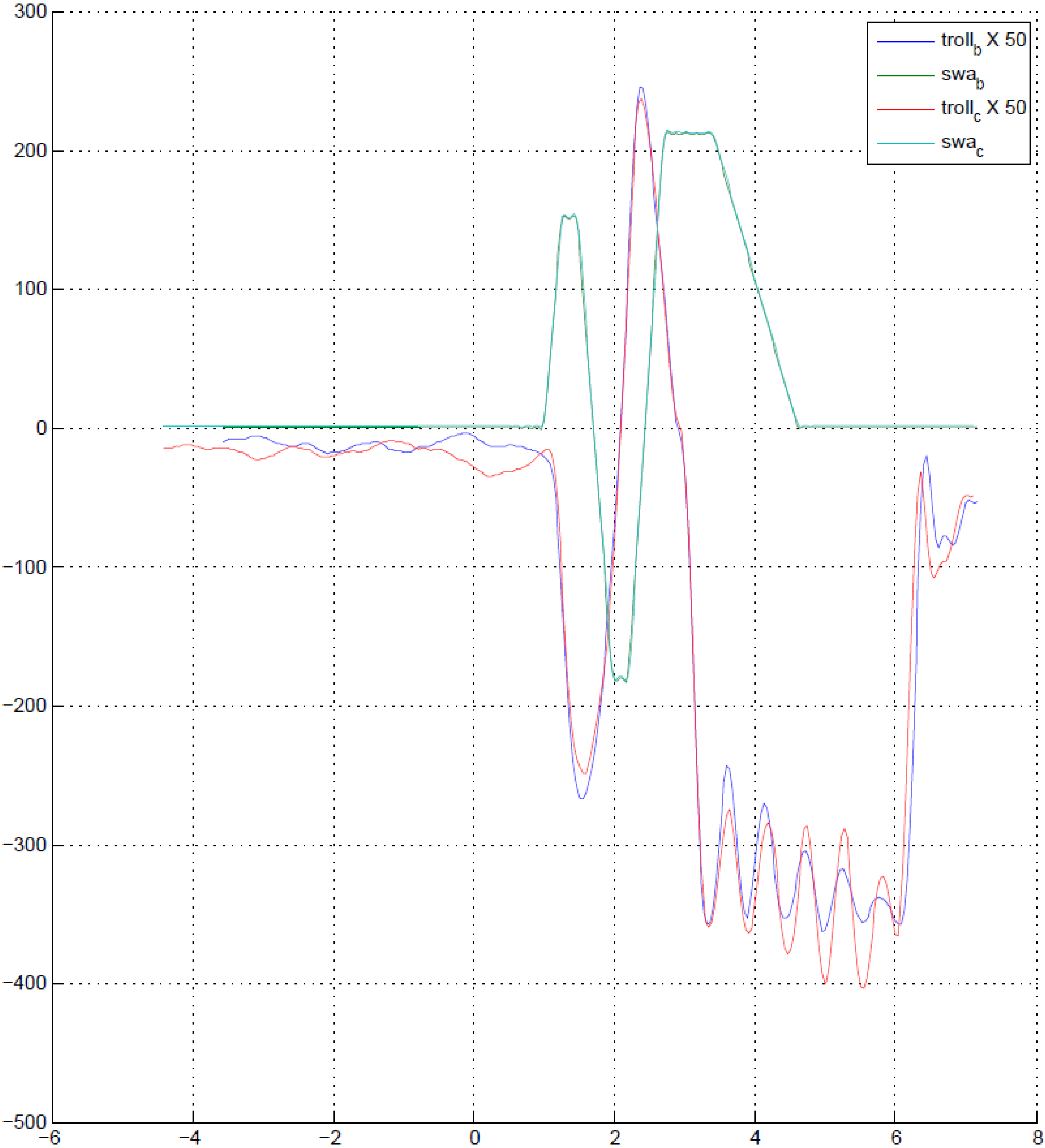
**Yaw / Lateral Acceleration vs Steering Wheel Angle with Moderate Steering Amplitude**
**Prior to Body Modifications**



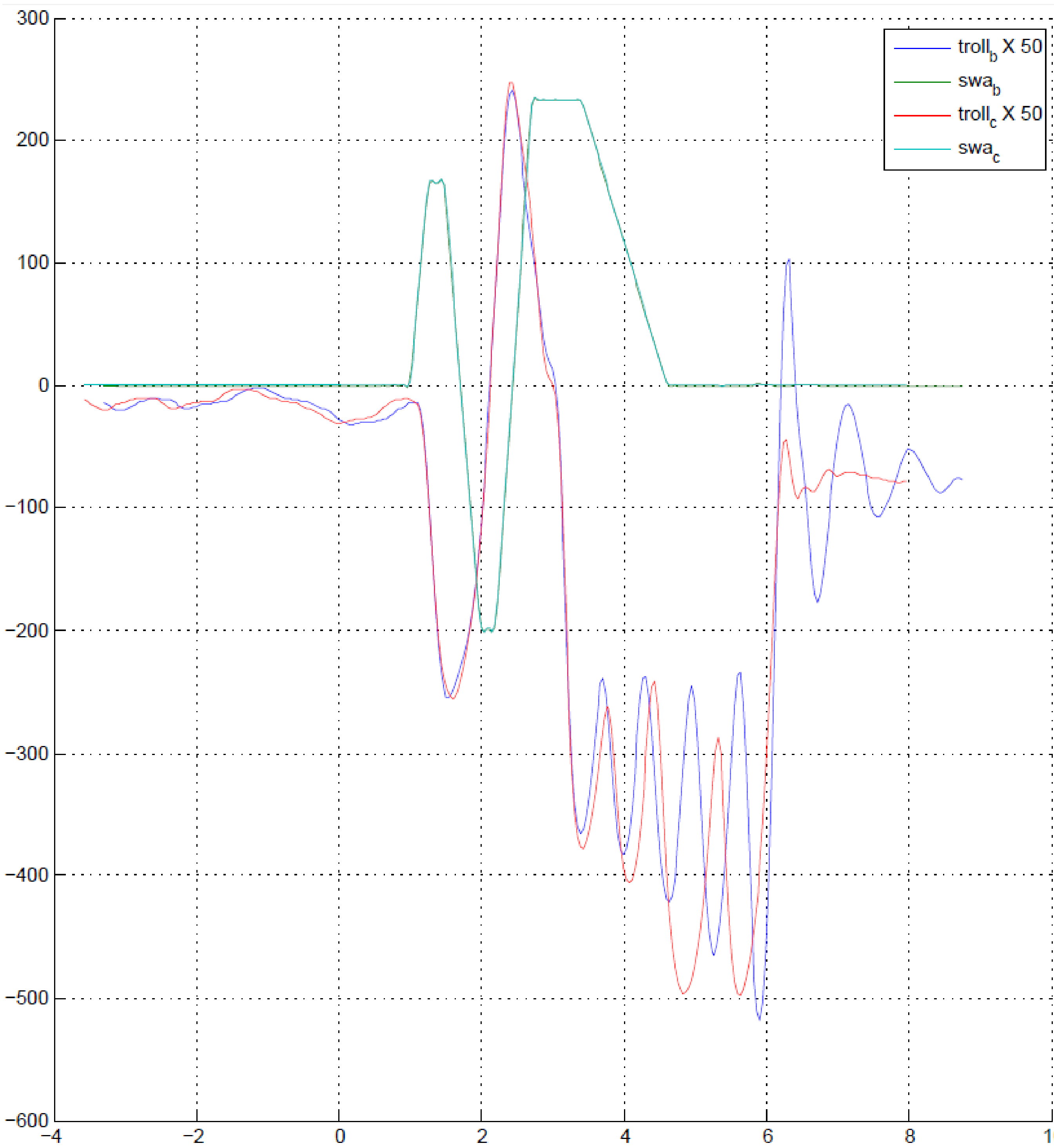
**Yaw / Lateral Acceleration vs Steering Wheel Angle with High Steering Amplitude Prior to Body Modifications**



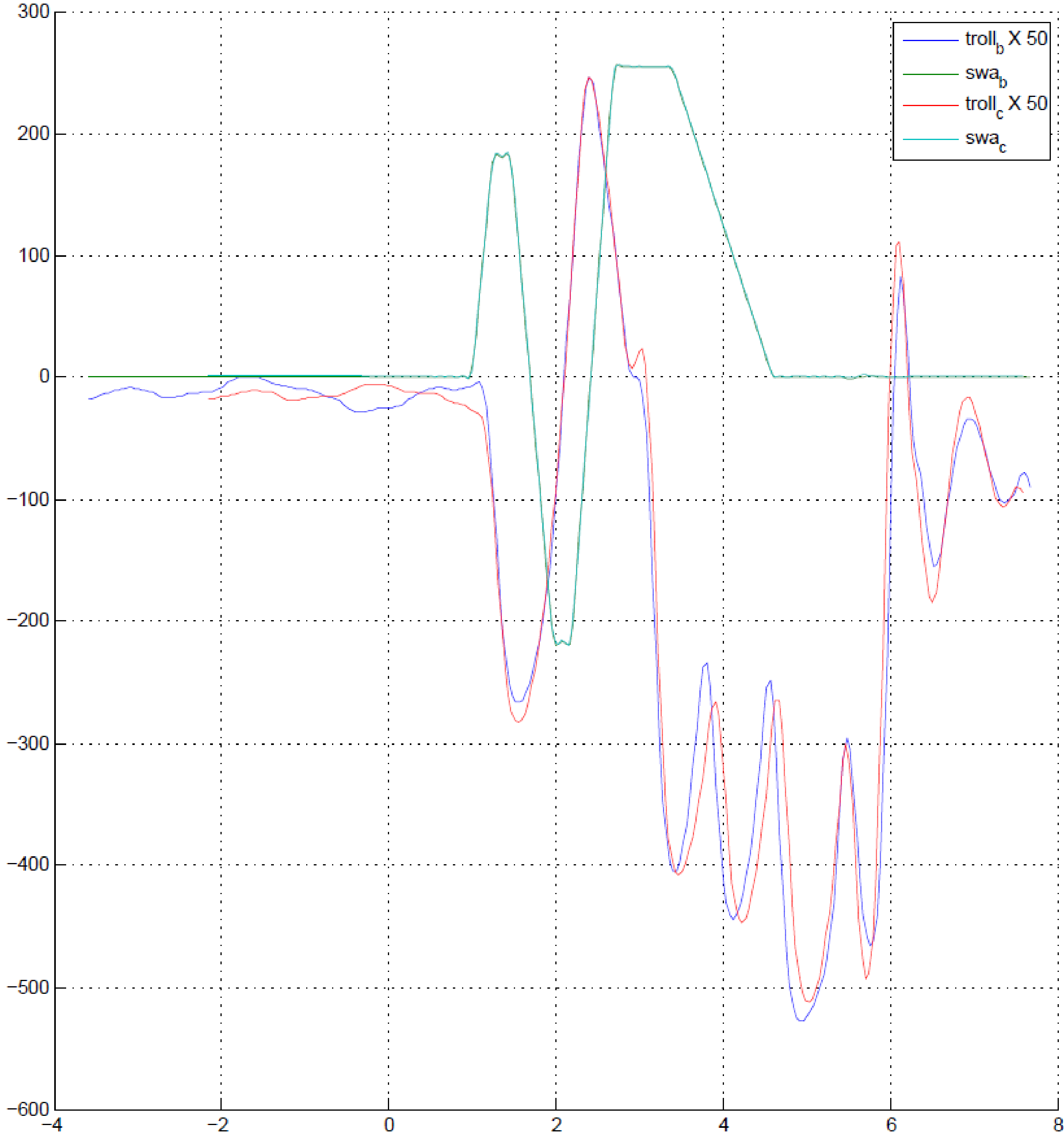
**Roll vs Steering Wheel Angle with Low Steering Amplitude**
**Prior to Body Modifications**



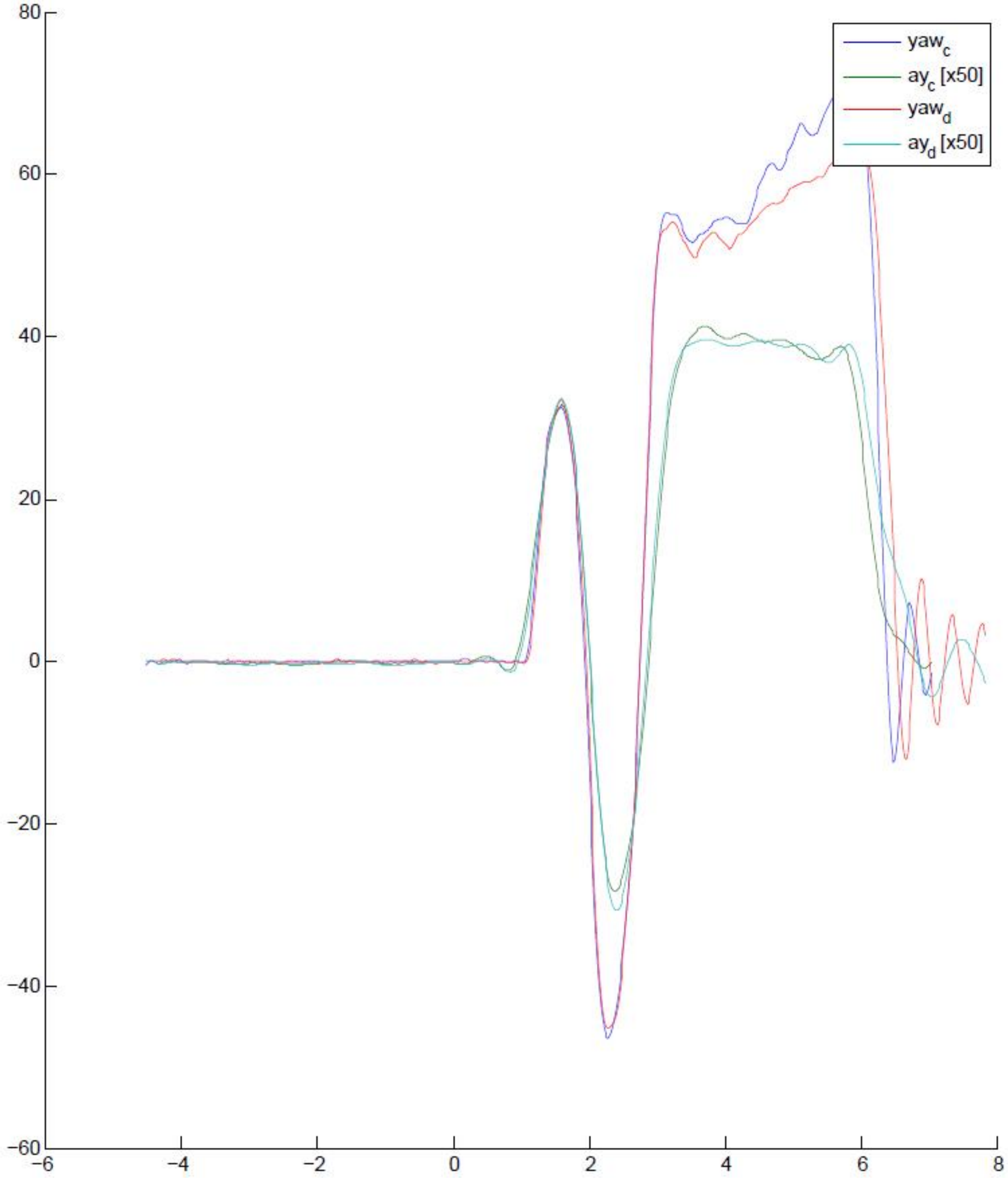
**Roll vs Steering Wheel Angle with Moderate Steering Amplitude Prior to Body Modifications**



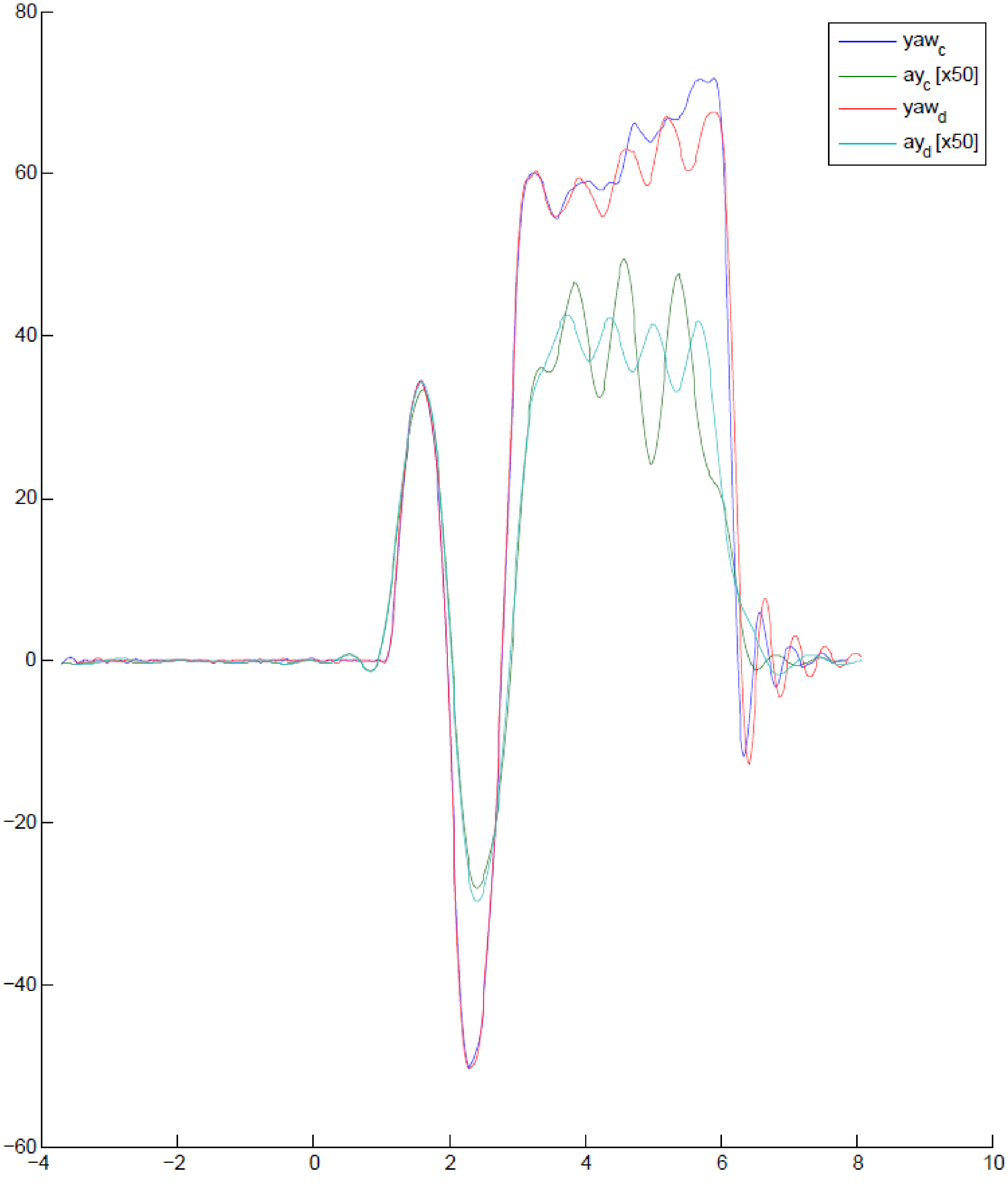
**Roll vs Steering Wheel Angle with High Steering Amplitude Prior to Body Modifications**



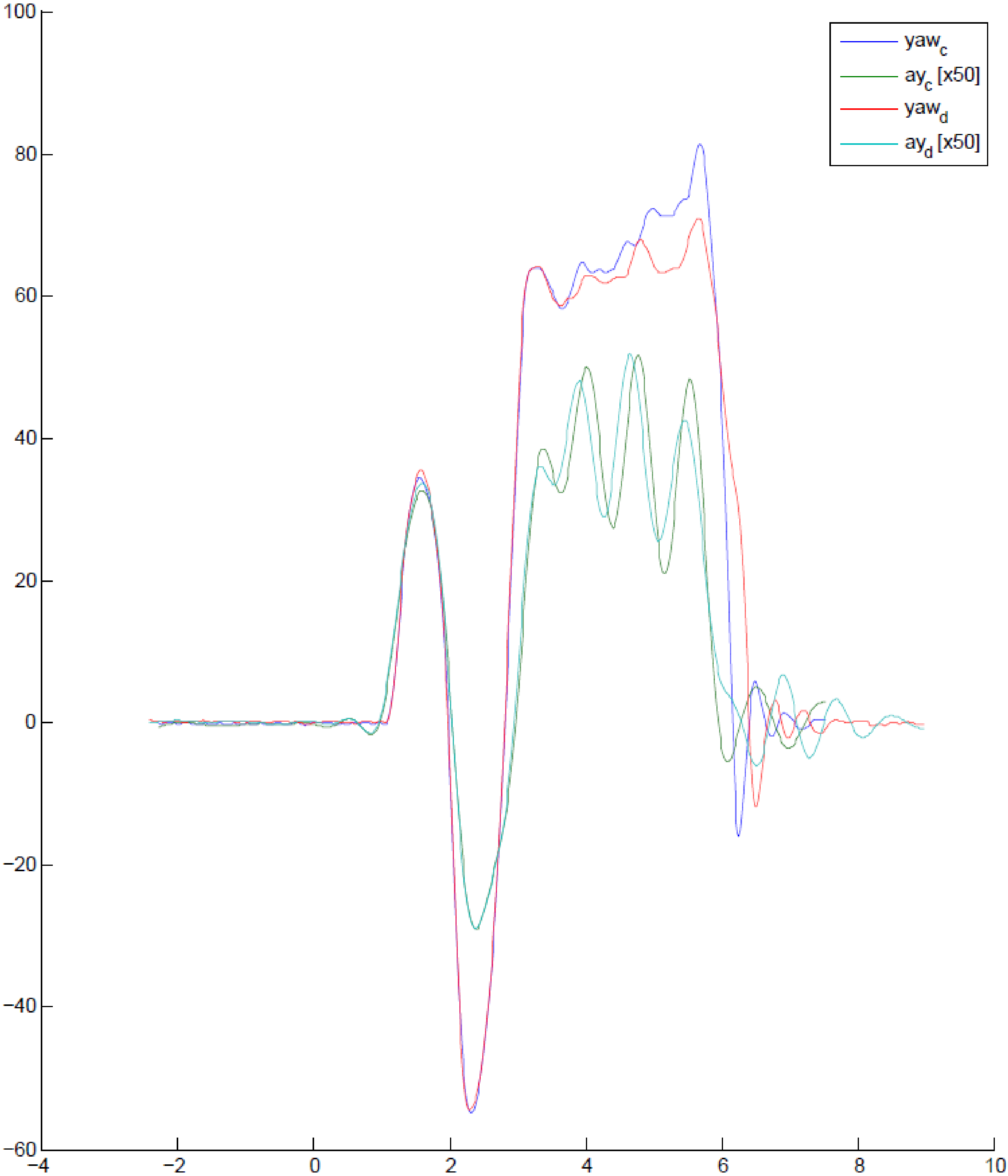
**Yaw / Lateral Acceleration vs Steering Wheel Angle with Low Steering Amplitude Post Body Modifications**



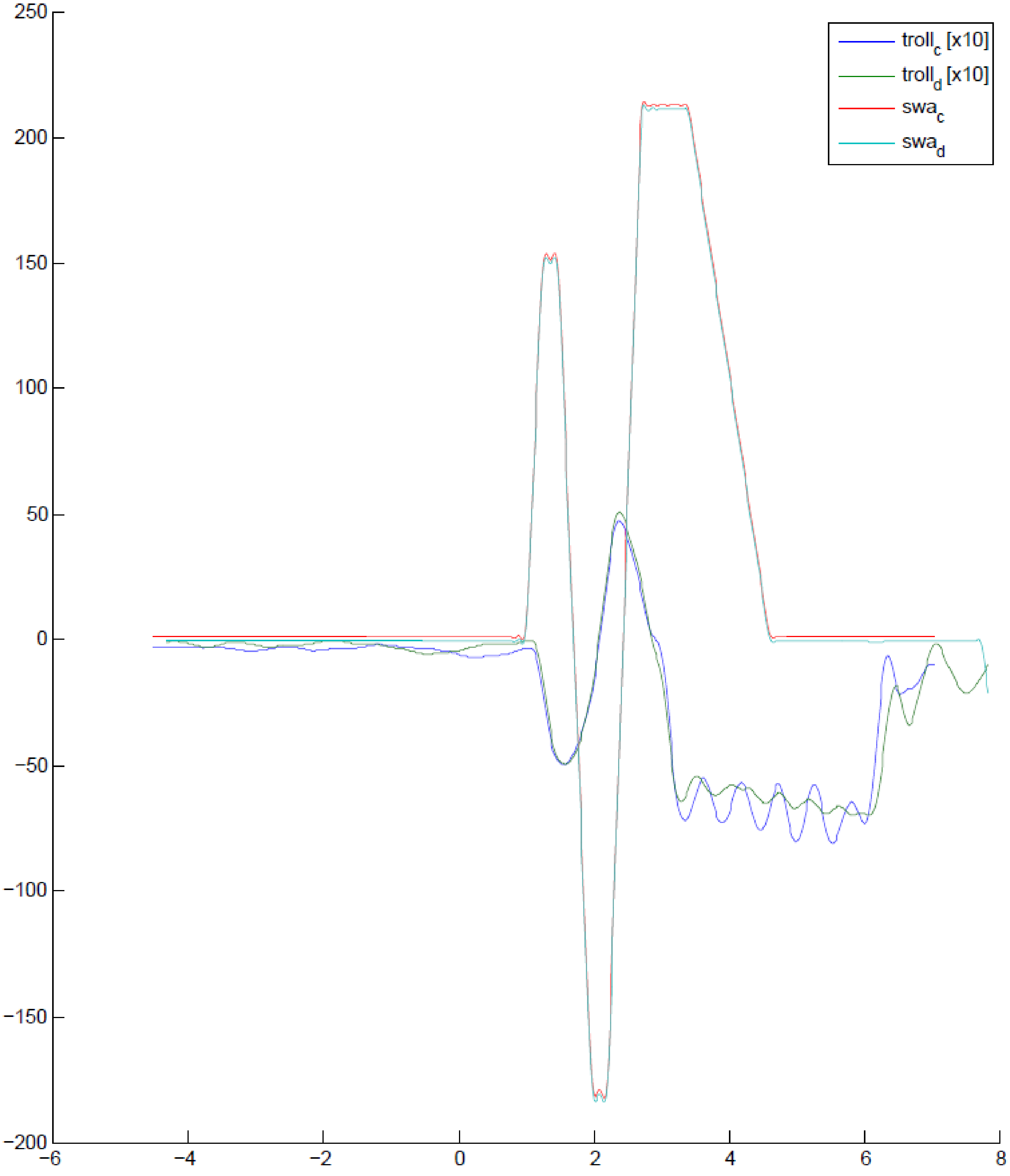
**Yaw / Lateral Acceleration vs Steering Wheel Angle with Moderate Steering Amplitude**
**Post Body Modifications**



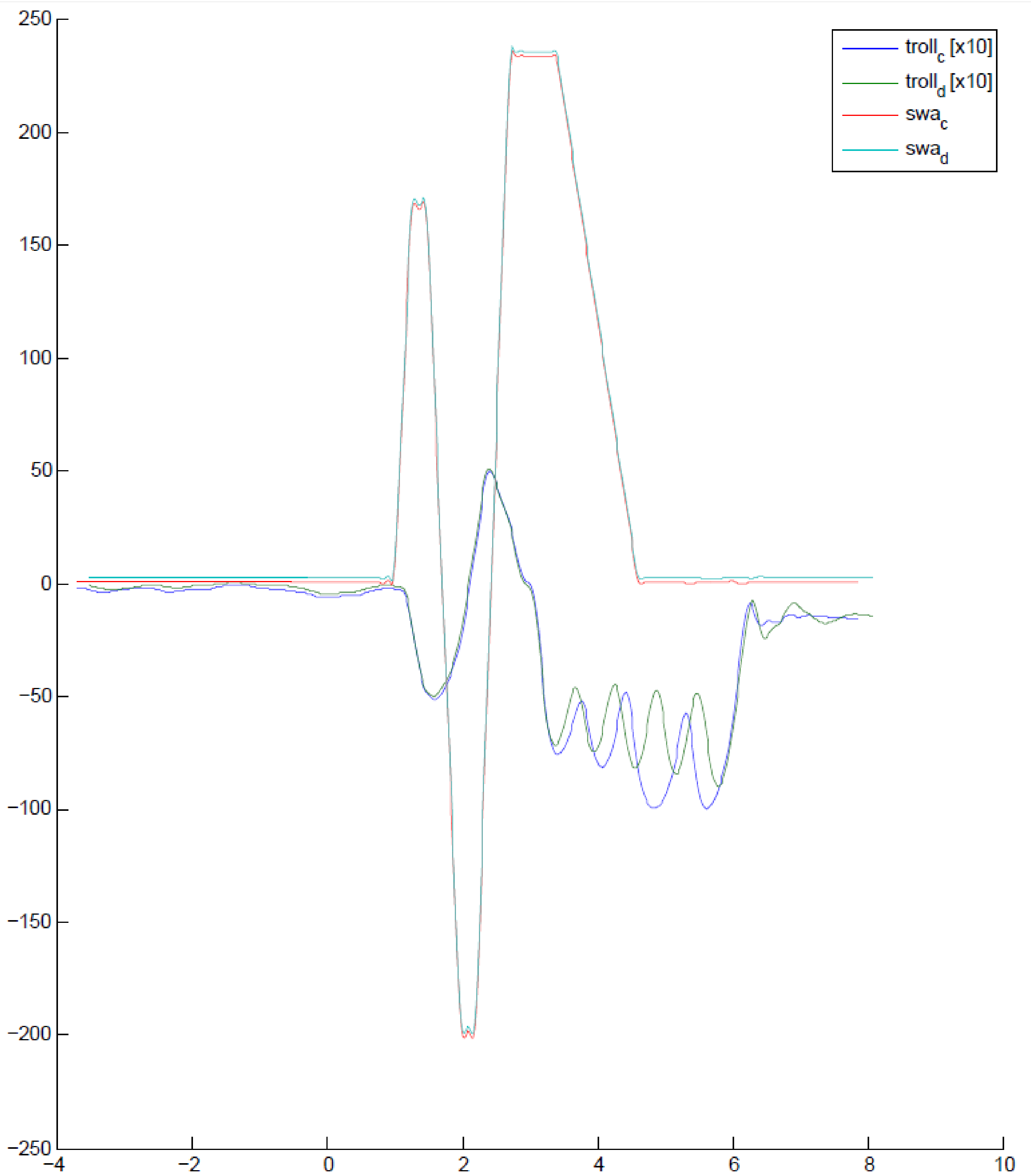
**Yaw / Lateral Acceleration vs Steering Wheel Angle with High Steering Amplitude**
**Post Body Modifications**



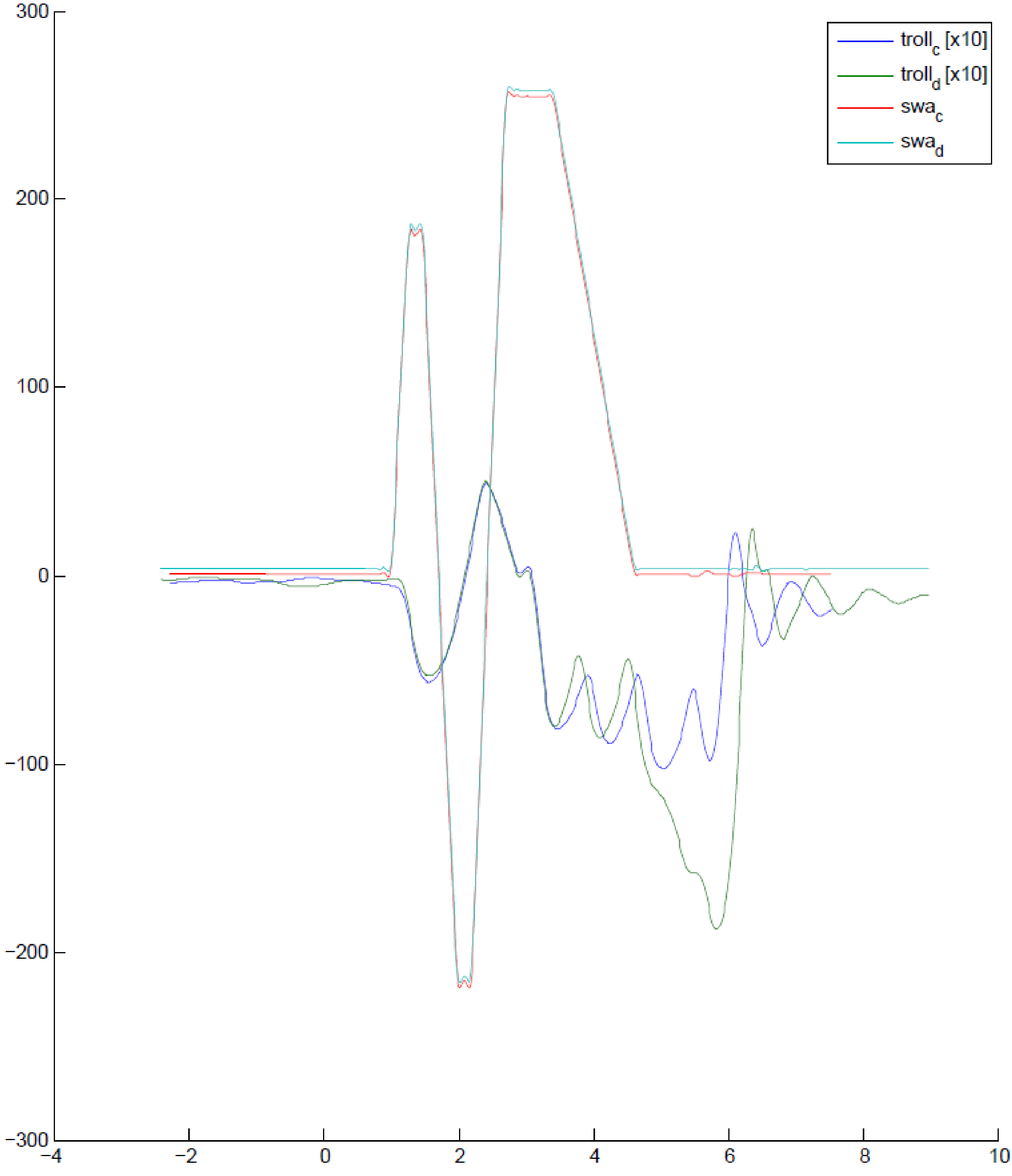
Roll vs Steering Wheel Angle with Low Steering Amplitude
Post Body Modifications



**Roll vs Steering Wheel Angle with Moderate Steering Amplitude**
**Post Body Modifications**



**Roll vs Steering Wheel Angle with High Steering Amplitude
Post Body Modifications**



# Stiffness investigation

Koen Bex 07/11/2014

### Vehicle & evaluation setup

Vehicle TG3S7: MWB LHD 330 Kombi 2.2L 100PS Diesel
Tyres 215/65R16C 109/107 Continental Vanco2 6YOL ET4E (OEM spec 2795)
Pressure: Front 3.5 bar Rear: 4.75 bar

### Vehicle evaluation as received

01/07/2014 Config: GVM-UDL
14u, Sunny Weights:
Track: 36°C
Amb:18.5°C

| | | |
|---|---|---|
| 785 | 753 | kg |
| 923 | 902 | |

Without bumper beam (vs with)

Steering: On center feel gone, no torque build up no V-shape. Delayed and very progressive response. -1.0 VER

Handling: No confidence feel due to poor steering. Roll control worse, less controllable lane change -1.2VER. Borderline acceptable for Trustmark.

### Vehicle evaluation with reinforcements

06/11/2014 Config: GVM-UDL
16u, Clouded Weights:
Track: 10°C
Amb:8.5°C

| | | |
|---|---|---|
| 783 | 752 | kg |
| 924 | 908 | |

Without bumper beam (vs with)

Steering: Slightly bigger dead window on center, a bit more torque build up (V-shape) on center. Similar response off center -0.2 VER

Handling: Similar performance, no real difference noticable.

### Conclusion

**Removing the rear bumper beam on the vehicle results in an on acceptable drop of VeD performance (Trustmark borderline). When removing the bumper beam on the reinforced vehicle, the drop in VeD performance is almost negligible (-0.2VER for steering)**